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STEAM TRAPS IN THE NAVAL SERVICE.

F. G. HECHLER, ASSOCIATE MEMBER.*

The steam trap is often considered a minor auxiliary requiring little care in its selection and even less for its operation. When once installed it is expected to successfully remove all condensation from steam lines, separators, heating systems, etc., and to do this without permitting the escape of any steam. To be considered successful the trap must be capable of doing this for long periods of time with practically no attention for repair or overhaul.

STEAM TRAP CLASSIFICATION.

Steam traps are usually divided into two broad groups, return traps and non-return traps. Return traps are so designed that they may be used to return the condensate directly to the boiler without the aid of a pump. For this purpose the traps are usually installed at a slightly higher level than the boiler into which they are to discharge, and they are so

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arranged that when the trap dumps it opens up a connection admitting steam directly from the boiler to the trap. This equalizes the pressures, and the water then drains by gravity from the trap back to the boiler. Return traps are not used in the Naval Service. Non-return traps discharge into the atmosphere or into a vessel where the pressure is not much greater than atmospheric, and hence do not need a steam equalizing valve because the pressure in the trap is in all cases higher than in the discharge line.

The non-return traps may be divided into the following classes: Open-bucket float, ball float, expansion, differential.

Of these four classes the first two are very extensively used for all classes of service; their outstanding characteristic is their prompt removal of all condensation as fast as formed. The expansion or thermostatic type is very popular for some classes of low-pressure work, such as heating systems; this type usually requires some cooling of the condensate before operating, which is desirable for this kind of work. Thermostatic traps are rather limited in their application, as they will not, in general, function satisfactorily over a wide pressure range without readjustment. The differential type is not used in the Naval Service.

The requirements for Naval Service are unusually severe compared with those of ordinary commercial use. The trap must function satisfactorily at all pressures from atmospheric to 300 pounds gage, the valve mechanism must be simple, all passages must be ample in size and not liable to become clogged with scale or sediment, and the discharge valve must not be at the bottom of the trap where scale and sediment collect. The valve mechanism should be readily accessible for inspection and adjustment or renewal without having to disconnect any pipe lines, and the body of the trap should be heavy and rigid so that it will safely withstand the severe service. Particular attention should be given to the design of the cover and the spacing and size of the cover bolts, or it will be difficult to keep the joint tight. The requirements of either

the Navy Standard or the American Standard for extra-heavy-flanged fittings should be complied with in these particulars. Comparatively wide gasket spaces are necessary at all joints; if they are too narrow it is difficult to keep the gaskets from blowing out. It is also easier to keep gaskets in place when counterbored recess-gasket flanges are used instead of raised faces.

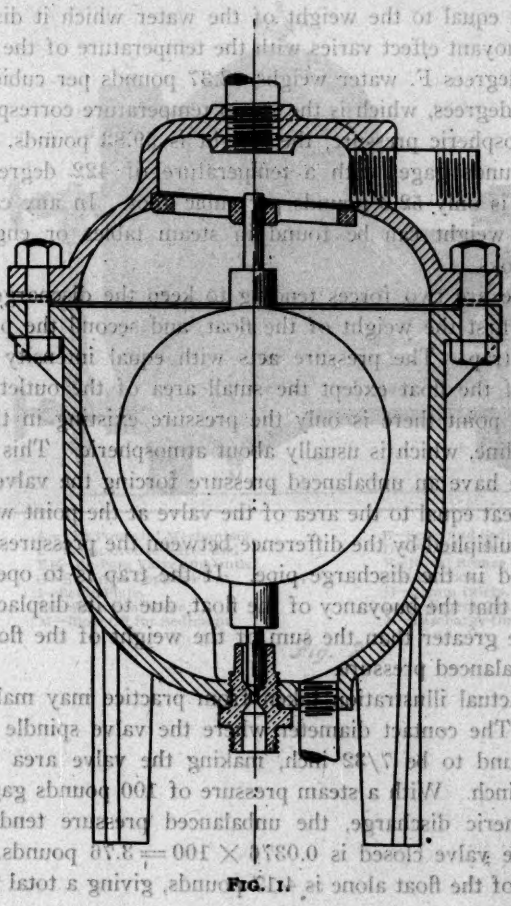


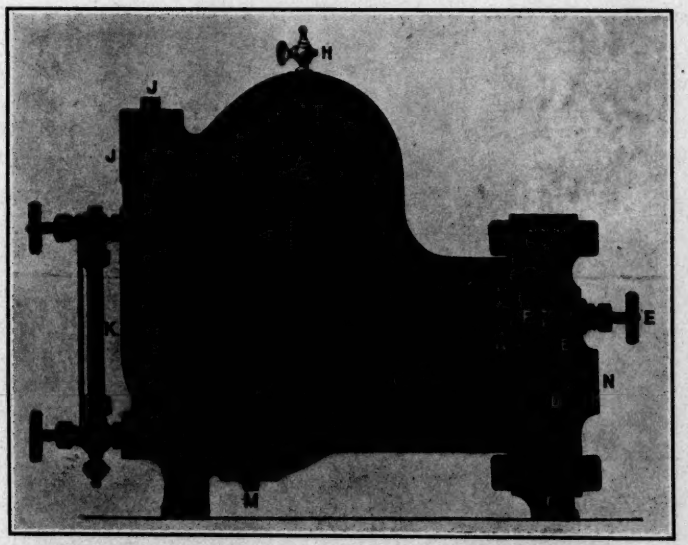
FIG. 11.

STEAM TRAP OPERATION.

Figure 1 illustrates a simple type of steam trap and is reproduced here to illustrate why trouble is sometimes experienced in getting steam traps to function properly under high pressures. In the design shown condensate enters the trap at the top and is discharged through the outlet valve at the bottom. As the water rises in the trap the float is buoyed up by an amount equal to the weight of the water which it displaces. This buoyant effect varies with the temperature of the water; at 60 degrees F. water weighs 62.37 pounds per cubic foot; at 212 degrees, which is the steam temperature corresponding to atmospheric pressure, the weight is 59.83 pounds, and at 300 pounds gage, with a temperature of 422 degrees, the weight is only 52.6 pounds per cubic foot. In any case the proper weight can be found in steam tables or engineers' handbooks.

There are two forces tending to keep the discharge valve closed, first the weight of the float, and second the pressure in the trap. The pressure acts with equal intensity on all parts of the float except the small area of the outlet valve. At this point there is only the pressure existing in the discharge line, which is usually about atmospheric. This means that we have an unbalanced pressure forcing the valve down on its seat equal to the area of the valve at the point where it seats multiplied by the difference between the pressures in the trap and in the discharge pipe. If the trap is to operate it is clear that the buoyancy of the float, due to its displacement, must be greater than the sum of the weight of the float and the unbalanced pressure.

An actual illustration taken from practice may make this clear. The contact diameter where the valve spindle seated was found to be 7/32 inch, making the valve area 0.0376 square inch. With a steam pressure of 100 pounds gage and atmospheric discharge, the unbalanced pressure tending to keep the valve closed is $0.0376 \times 100 = 3.76$ pounds. The weight of the float alone is 4.17 pounds, giving a total weight

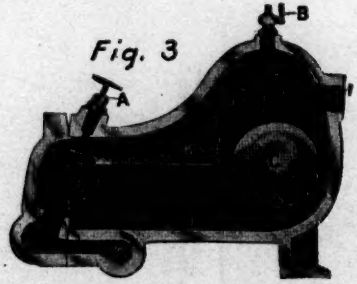


A—The Powerful Mechanism.
C—Tobin Bronze Valve Rod.
EE—By-Pass Valve and Outlet.
G—Seamless Copper Float.
I—Baffle Plate.
K—Water Seal Gage.
M—Blow Off for Sediment.

B—Scale and Sediment Strainer.
D—Phospher Bronze Valve Seat.
F—Naval Bronze Seat.
H—Air Cock.
JJ—Steam Inlets.
L—Sediment Chamber.
N—Discharge Outlet.

Fig. 2

Fig. 3



A—Wheel valve to depress lever, opening valve.
B—Air Valve.
C—Seamless copper float, extra heavy.
E—Lever joint.
F—Valve stem.
G—Valve seat.
H—Outlet.
I—Inlet.
J—Sediment discharge.
K—Lever.

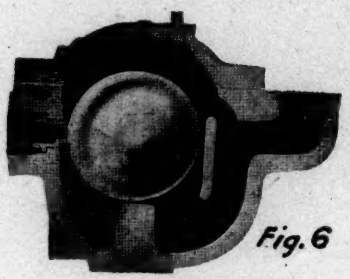


Fig. 6

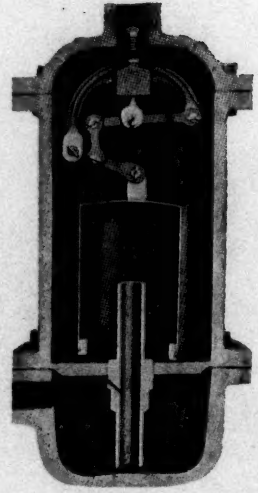


Fig. 7

Fig. 8

of 7.93 pounds. To overcome this downward force we have the buoyancy of the float. By computation its displacement, when fully submerged, was found to be 221 cubic inches, and as the weight of water per cubic foot at the temperature corresponding to 100 pounds gage pressure is about 56 pounds, the buoyancy of the float is only 7.2 pounds, hence the force tending to open the valve is less than that keeping it closed, and the trap will not operate. If the pressure is decreased to about 70 pounds the unbalanced pressure is sufficiently reduced, and the buoyancy at the same time increased, due to the change in density of the displaced water, that the two forces balance and the trap discharges.

The trap with the dimensions given was submitted for test up to 300 pounds gage. In order to make it operate at this pressure the size of the outlet valve would have to be greatly reduced. This size could be found as follows: Buoyancy of float at 300 pounds $= 221 \div 1728 \times 52.6 = 6.72$ pounds. The weight of the float is 4.17 pounds. Hence the maximum unbalanced pressure cannot exceed $6.72 - 4.17 = 2.55$ pounds. Hence, the contact area of the valve must not be greater than $2.55 \div 300 = 0.0085$ square inch or about ¼ of the original area. The corresponding diameter is slightly less than 7/64 of an inch. Such a large decrease in the size of the outlet reduces the capacity of the trap to a prohibitively small value and also increases the liability of scale and sediment lodging in the passages.

Two methods are available for increasing the capacity of the trap. First, a larger float may be used, thus increasing the buoyancy faster than the weight. The principal objections to this are the difficulty of building a large float that will satisfactorily withstand the high pressures and the large increase in the size and weight of the trap. A better and more common solution is the use of a compound-lever system for magnifying the power of the float or bucket. Such a solution as applied to float traps is shown in figures 2 and 3. The same scheme as applied to bucket traps is shown in figures 4

and 5. By properly choosing the ratio of the levers the pulling power of the float or bucket can be greatly increased. However, to secure this increased power it is necessary to considerably increase the travel of the float or bucket in order to move the valve sufficiently far from its seat to permit a free discharge.

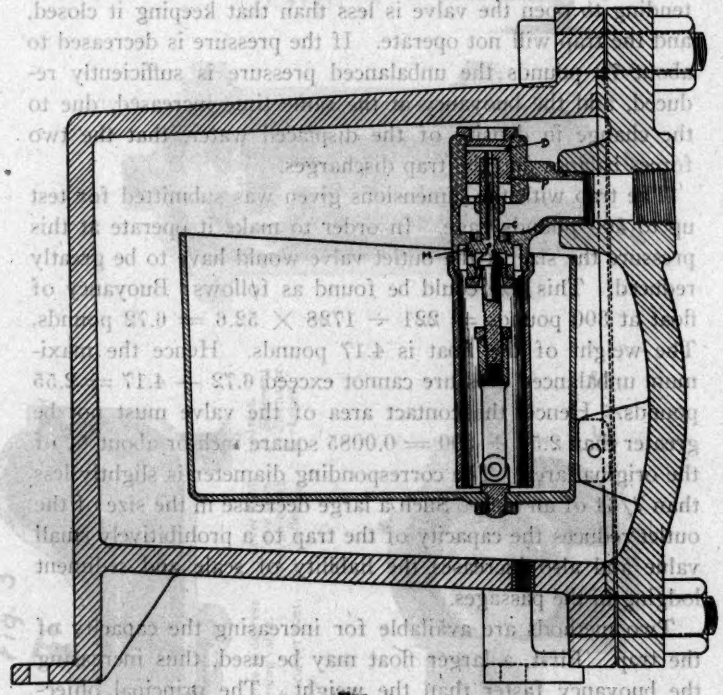


FIG. 4.

An examination of figure 4 will show that a double valve system is used, consisting of a control or pilot valve C, and the main discharge valve M. As the water level rises in the trap the bucket floats and keeps both valves on their seats until the water rises high enough to overflow into the bucket. As the bucket fills, it loses its buoyancy and sinks. Its weight,

acting through the leverage, is sufficient to open the small control valve C. This admits steam pressure to the chamber D on top of the piston P, which is attached to the main valve M. The piston P and valve M have the same area, so that when the pilot valve opens and admits steam to chamber D the main valve is balanced and opens readily as the bucket continues to descend. This arrangement permits the use of a large main valve, giving the trap a large capacity. Some method should be provided to permit the escape of the water trapped in the chamber D, otherwise it may prevent the valves from closing promptly.

There are a number of bucket traps similar to the trap shown in figure 4, in which the compound-valve system is not used. In these designs the necessary operating power is obtained by making the distance between the bucket fulcrum and the point of attachment of the valve stem as short as possible, and also by properly fixing the size of the discharge valve. To secure a short connection the end of the bucket next to the hinge is often made inclined, so that when the bucket drops this edge will be close to and parallel with the discharge tube.

The operating power available for opening a valve against the steam pressure is readily computed in the case of a float trap from the proportions of the various levers, taking the point of application of the buoyant force of the float to be at its center. With bucket traps it is not quite so easy to find the result by computation unless the bucket is symmetrical and of uniform thickness. It is usually simpler to attach the bucket in its operating position, and fill the trap with water so that the bucket will be submerged. In the case of the trap shown in figure 4 it will be necessary to remove the head and bucket and submerge them in a pail of water. Then attach a spring balance to the bucket at the point of attachment of the valve and find the pull required to lift the bucket. This gives the force available for opening the discharge valve. This result is only slightly influenced by the temperature of the water because the volume displaced by the metal in the bucket

is quite small. The closing force is not the same, but is dependent on the buoyancy of the bucket, which is equal to the weight of water displaced, and is considerably affected by the temperature of the condensate. In every case the opening force should be greater than the area of the valve, measured on its seating diameter, multiplied by the steam pressure.

CAPACITY OF STEAM TRAPS.

From the foregoing discussion regarding the necessary relation between valve size and available operating power of float or bucket, it is evident that steam traps will have widely varying capacities, depending on the design. It is customary to designate steam traps according to the size of the inlet and outlet pipe connections. This method has nothing but custom to recommend it. Traps of the same nominal size often have widely different capacities. For instance, with the 1¼-inch traps tested at the Naval Engineering Experiment Station, the minimum capacity of any trap at 250 pounds steam pressure was found to be 2,300 pounds of water per hour, while the maximum was 26,300 pounds, or more than eleven times as much. The latter trap had a compound-valve arrangement similar to that of figure 4. A capacity rating of a steam trap is of no value unless the pressure at which it was made is also given; often the published ratings represent the capacities at the maximum pressure for which the valve is designed.

In order to secure satisfactory capacity results from steam traps it is desirable that they should be provided with interchangeable valves and valve seats adapted to various pressures. The number of valves thus provided is usually limited to three, covering certain pressure ranges. One should be a low-pressure valve for use from atmospheric pressure up to about 30 pounds gage; one an intermediate valve for pressures from 30 to 150 pounds, and one a high-pressure valve for pressures from 150 to 300 pounds. The high-pressure valve will, of course, operate at any pressure down to atmospheric, but with a greatly reduced capacity, because of its small size. Thus

a 1¼-inch trap which discharged 3,200 pounds of water at 250 pounds pressure discharged only 900 pounds with the same valve and a steam pressure of 10 pounds. By changing the valve to the proper size for this pressure the capacity of the trap at 10 pounds pressure was increased to 4,700 pounds. All valves should operate satisfactorily up to pressures somewhat higher than their rated value; this means that a 30-pound valve should operate up to say 40 or 45 pounds for a margin of safety. This also makes it possible to grind in the valves when they become worn or even to ream out the seat. Both of these operations are apt to enlarge the seating area of the valve with the consequent danger of the trap becoming inoperative at the higher pressures. This sometimes makes it necessary to replace valves and seats with new ones of the proper size.

A COMPARISON OF OPEN-BUCKET AND BALL-FLOAT TRAPS.

There does not seem to be much choice between these two types of traps for most kinds of service, and both types are giving satisfactory results. The bucket trap is of the intermittent-discharge type, the bucket alternately filling and dumping. The action when opening and closing should be sharp and positive, and when in the closed position there should be no dribbling of steam or condensate. The ball-float traps, on the contrary, discharge continually so long as condensate flows to them, and when working properly they will maintain an absolutely constant water level. It is often claimed that the constant wire-drawing action in the constant-discharge type is destructive to the valves and seats, and that in this respect the intermittent-discharge type is preferable. On the other hand, the constant opening and closing of the valve of the intermittent type will eventually wear a groove in the valve, especially when sharp-edged seats are used.

When first put in service the open-bucket type will have its discharge valve wide open, and it will remain so until enough condensate enters to buoy up the bucket. Usually this will be no disadvantage since air, followed by condensate,

usually enters the trap ahead of steam. This gives the air a good chance to escape. If a trap is blown down, however, and then dry steam only enters, the trap may continue to blow steam for some time. Ball-float traps are always closed when empty, and the discharge valve cannot open until the water level rises to the desired height.

ACTION OF STEAM TRAPS ON A VESSEL.

When installed on a vessel in a heavy seaway the operation of both types is interfered with. In stationary service traps are always installed in a horizontal position, but on a vessel their position may vary widely from moment to moment. Traps having attached buckets or floats are most sensitive to oscillations about an axis at right angles to a plane containing the bucket and the lever arms. Whenever a certain angle of inclination is exceeded in one direction the discharge valve will usually open and permit steam to escape. This angle is usually found to be about 20 to 25 degrees; it is dependent on the depth of the water seal normally provided for the discharge valve and on the distance between the float and its pivot. The shorter this distance can be made the better the results will be. Tilting the trap in the opposite direction has no effect on its functioning. Also when the trap is rocked sideways about an axis in the plane of the bucket and the lever arms there is no interference with its operation. Consideration of this result would indicate that these traps should be installed in a fore-and-aft direction whenever convenient to do so. Not all traps, however, are adversely affected by the rolling of the ship. Tests made on the two types shown in figures 6 and 7 show that the operation of these two designs is not affected by any angle of inclination up to 35 degrees, no matter in what direction the trap is installed.

STEAM TRAP FITTINGS.

There are certain auxiliary fittings often placed on steam traps as an aid in maintaining and determining their condition of operation. These fittings include a bottom blow for clear-

ing the trap of any accumulation of scale or sediment, an air cock for relieving the air at intervals to prevent air binding, a gage glass so that the operation of the trap may be seen at a glance, a hand gear for operating the valve to determine whether or not it is sticking, and an accessible and removable strainer for trapping scale and sediment before it reaches the trap.

Of these aids in steam-trap operation the gage glass is the hardest to keep in satisfactory condition. The ordinary tubular glasses are not reliable at high steam pressures and often require frequent renewal. For high steam pressures flat glasses of the Klinger, or reflex type, are much more satisfactory. A little device, figure 8, that has given excellent results as a gage-glass substitute is shown in the accompanying illustration. This device, known as a thermo gage, is attached to the trap just below the normal water level that it is desired to maintain. At the upper end is a disc radiator and a pet cock. So long as the trap operates properly the gage will be filled with hot water, and the radiator discs will be found to be progressively warmer from the top down. If the trap becomes water logged the radiator becomes cold, while if it leaks the water seal in the gage is lost and the gage has a temperature approaching that of the steam. The convenience of the device is apparent, as it needs only a touch of the hand to determine how the trap is operating.

TRAP INSTALLATION.

Steam traps should always be placed lower than the lowest line they are intended to drain, and preferably above the level of the tank into which they discharge. No trap should be connected to drain more than one line unless the pressures are sure to be identical. They should, whenever possible, be located in an accessible position for convenience of inspection and repair. Other things being equal, the longitudinal axis of the trap should be placed fore and aft, rather than athwartships, because the operation of the trap will be better in a heavy seaway. Traps should always be connected with unions

to facilitate removal, and a by-pass should be provided for use when the trap is out of service. The bottom blow-off should be piped to the bilge.

TYPES OF TRAPS.

The external appearance of the traps shown in figures 2 and 3 is quite similar and both belong to the ball-float, constant-discharge type. The valve arrangement and the connection between the float and the valve is, however, quite different in the two designs. In figure 2 the float operates the valve through a powerful toggle-joint arrangement, insuring positive control of the valve. The valve is placed well up above the bottom of the trap where it will be free from sediment and scale, and as an additional safeguard a strainer is fitted on the valve mechanism. The strainer, however, is not accessible without dismantling the trap, and, furthermore, the inlet and outlet connections, being located in the body and the cover respectively, make it necessary to break the pipe line before the trap can be opened up.

In the design shown in figure 3 the valve opens with the pressure instead of against it as is usual. In this case the float and attached parts must be heavy enough to keep the valve closed at the highest pressure and yet the buoyancy must be sufficient to open the valve when the water level in the trap rises to the desired point. With this arrangement the danger of fouling the valve is reduced to a minimum. The handwheel A is so arranged that when it is screwed in against the lever the valve is opened, clearing the trap and cleaning the seat. The inlet and outlet connections are placed in the body of the trap, making dismantling easy. It will be noted that all of the operating parts are attached to the cover which may be taken to a work bench and any needed adjustments made conveniently.

Figure 4 is representative of a type very largely used in the Naval Service. With this type either the single or compound-discharge valve may be used; the latter gives the trap a very large capacity. This trap has the inlet and outlet

connections in the cover, so that it can be readily dismantled provided the body is not fastened to the floor and there is sufficient space to clear the bucket. The counterbored-recess type of gasket flange as used on this trap is excellent. Best results will be obtained by carefully fitting the male and female ends so that there is only a small clearance, otherwise the gasket may be blown out.

Figure 5 shows another type of open-bucket trap used to some extent in the Naval Service. A compound valve arrangement differing only in details from that already described is used. As the water level rises the bucket fills and drops, moving about the swivel joint G as a pivot. The float arm Q is fitted with a projecting fork which engages the nut L on the pilot-valve stem O, thus lifting the pilot valve when the bucket sinks. The condensate is then forced out through H, Q, G and the discharge valve to the outlet. This particular design requires a comparatively large trap body, the side walls of which are usually made flat surfaces. Such a construction is not satisfactory for high-pressure work unless carefully reinforced, and then the weight and space required are excessive in proportion to the capacity.

What is probably the simplest trap suitable for high-pressure work in use at present is shown in figure 6. It consists of only four parts: body, cover, float and outlet valve. The ball is the only moving part, and it is in no way restrained. When not under pressure the ball rests on the lug at the bottom of the trap. As soon as the condensate enters, the ball floats and is forced against the valve seat by the unbalanced steam pressure. As the water level continues to rise the buoyancy of the ball overcomes the force holding it on the seat, and the float rolls up about the top edge of the valve as a fulcrum, thus permitting the condensate to discharge. Air, being heavier than steam and lighter than water, will form a layer between the two and be discharged first. This obviates all danger of the trap becoming air bound. This trap does not permit steam to blow through when oscillated up to 35

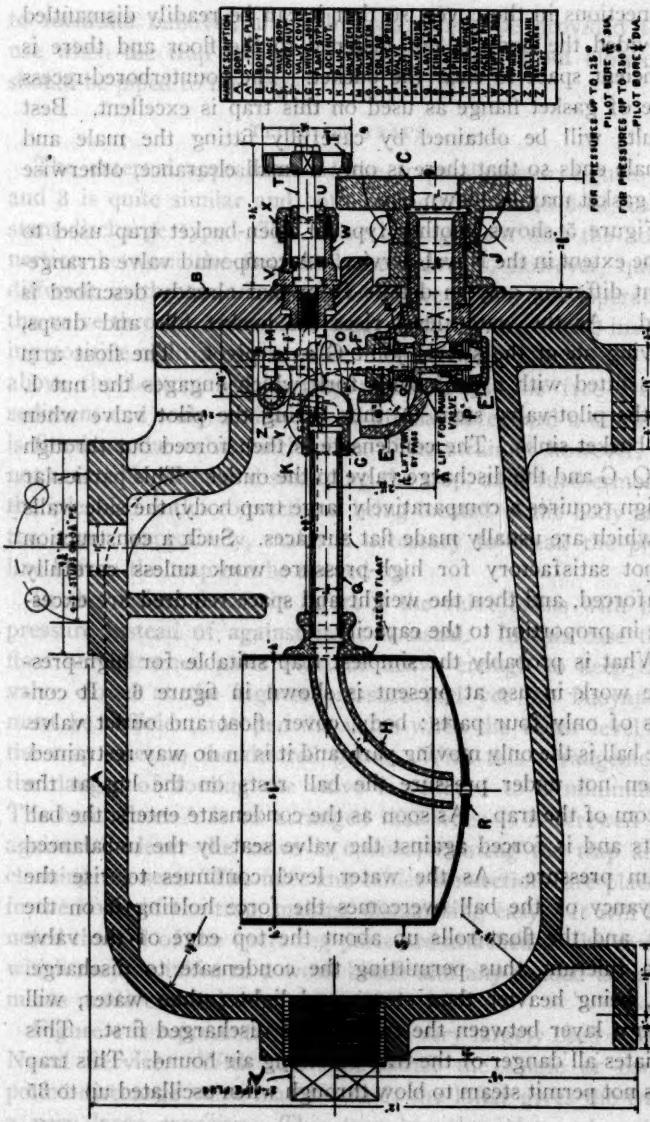


FIG. 5.

degrees from the horizontal position, that is through a total angle of 70 degrees. The ball floats used at this station were originally nickel plated. This plating has a tendency to flake off and may cause a slight unevenness of the surface. Care must be exercised when assembling the trap not to drop the ball into place or a flat spot may result. If this occurs a drop or two of solder at this point will make this side heavy and will prevent this spot from coming in contact with the valve and preventing proper closing. Floats for use with this trap when tested hydrostatically failed at about 1,000 pounds pressure, which indicates that they are amply strong for the service intended. The addition of a good strainer at the inlet would be desirable.

In figure 7 is shown a trap which operates on an entirely different principle from any of those previously shown. The inlet is at the left of the base and the outlet in the cover, so that the trap does not fulfill the ideal requirement that the design should be such that the trap may be opened up without disconnecting any piping. The operation of this trap is as follows: Condensate enters the base or separating chamber and passes up through the connecting brass tube into the upper or bucket chamber. In the position shown the bucket is in its lowest position and the outlet valve is open for discharge. As soon as all of the water is discharged from the base, steam enters and fills the bucket. This makes the bucket buoyant, since it is surrounded by water on the outside, and it rises, closing the valve. The steam in the bucket is constantly condensing, and in order to replenish it a small angular hole is drilled in the brass tube at the top of the base chamber. When the base chamber fills with condensate this passage becomes water sealed, and the bucket soon loses its buoyancy and sinks, opening the discharge valve. Any air entering the trap will find its way into the upper part of the inverted bucket, and if it were permitted to remain there the trap would soon become air bound and cease operating. To prevent this a very small hole is drilled through the top of the

bucket, this allows the air to escape into the top of the trap where it will be promptly discharged as soon as the trap opens. This trap is not adversely affected by being oscillated and does not need an air cock, since it cannot become air bound.

SUMMARY.

The desirable features of a satisfactory steam trap for use in the Naval Service may be summarized as follows:

Stong, rugged construction with properly designed gasket joints which may be kept tight without difficulty.

Inlet and outlet connections in the body of the trap.

The outlet valve located above the bottom of the trap where it is not likely to become clogged.

An accessible inlet strainer.

Powerful valve-operating mechanism so that the outlet valve may be large enough to give the trap ample capacity.

Proper functioning when oscillated.

A gage glass or other device for easily determining whether the trap is operating properly. For this purpose a hand gear for opening the outlet valve is also often convenient.

To secure the best results with traps they should be installed on the ship with their axis fore and aft to minimize the effects of a heavy sea. They should always be connected to the lines with unions, and every trap should be by-passed.

A FIFTY YEAR RETROSPECT OF NAVAL MARINE ENGINEERING.

By Rear Admiral C. W. Dyson, U. S. Navy, Member.

The armed forces of democracies ride on the crest of the wave of popular approval only in times of national peril. The peril past, the Army and the Navy are forgotten, save by a few, and these forces are practically compelled to fight, if not actually for existence, at least for sufficient support to maintain a nucleus about which efficient forces may be again built, *if time permits*, when another national emergency arrives.

At the time of my entrance into the U. S. Naval Academy as a Cadet Engineer, October 1, 1879, the Civil War was buried under fourteen years of peace, during which time the Nation had been taken up with the mighty problems of reconstruction of the Union, and of the rebuilding of industries and of private fortunes which had been wrecked in the turmoil of the strife. The Navy, as well as the Army, was withdrawn from the spotlight of the public stage and, being hidden in obscurity, was forgotten.

The vessels on the active list of the Navy were mostly relics of the period antecedent to the war and of the war period, and included among them one steam frigate fitted with side wheels. The building ways at the navy yards were occupied by wooden hulls laid down during and immediately subsequent to the war, but which, due to lack of appropriations to complete, were far advanced in decay at the period of which I write. At several of the ship yards of the country were the iron hulls of five monitors, some of which had already been launched, but all of which were waiting for turrets and machinery, which were not forthcoming on account of lack of money.

In order to maintain the small number of ships we had on the Navy list it was necessary, in some cases, to rebuild the vessels around a small piece of the original keel, as Congress would approve appropriations for repairs, but practically none for new construction, and therefore this rebuilding was effected under the head of necessary repairs.

Some small amount of new construction had been carried on, however, and in addition to the few *ante bellum* and *bellum* craft there were four wooden steam sloops, three steel steam sloops, and one wooden steam frigate, the *Trenton*, on the Navy list, which only dated back to 1876 and '77.

Congress, notwithstanding its general neglect of the Navy, had passed one piece of legislation of great value to the Naval Service. As was but natural, the rapid decay of *materiel* had been accompanied by as rapid a deterioration of the *personnel* of the Naval Service. From the general discouragement and carelessness which the existing conditions engendered, some few officers of all branches had held themselves aloof and had persevered against heavy odds in their attempts to better the condition of affairs and to keep themselves abreast of the times, trusting that the opportunity for using their talents would arise in the not distant future. Congress, by an act passed in 1864, had authorized the formation of classes at the Naval Academy for the study of Naval Architecture and Marine Engineering, and had designated such students as Cadet Engineers. The first results were very discouraging and the scheme was dropped in 1868, but was revived again in 1871, when a two years' course was established. This was again modified in 1874, when the course was extended to four years. The examinations for entrance were competitive, twenty-five new men being admitted annually. The system was short-lived, being legislated out of existence in 1882, when the Cadet Engineer and Cadet Midshipman classes were combined and the titles changed to Naval Cadet.

Brief though the time, and few the number of classes that

had been admitted, three classes of two years and eight classes of four years, the results as measured by the material obtained have probably never been surpassed by any educational institution. Among the graduates of these classes were the late Prof. H. W. Spangler, of the University of Pennsylvania; Prof. Mortimer L. Cooley, of the University of Michigan; Prof. Ira L. Hollis, president of Worcester Polytechnic Institute; Prof. W. L. Durand, late of Cornell University; Prof. W. H. P. Creighton, late of Purdue College, men who may be rated among the founders of modern marine engineering education in this country. Of those graduates still in the Naval Service but few now remain on active duty, but among them are to be found Rear Admiral R. S. Griffin, Chief of Bureau of Steam Engineering; Chief Constructor D. W. Taylor, Chief of Bureau of Construction and Repair; Rear Admiral Joseph Strauss, late Chief of Bureau of Ordnance, and Rear Admiral George E. Burd, Industrial Manager of the New York Navy Yard. Among those in civil life who have distinguished themselves are Mr. F. T. Bowles, late President of the Fore River Shipbuilding Company, and now Assistant General Manager of the Emergency Fleet Corporation; Mr. W. H. Gartley, Engineer of Works of the Philadelphia Gas Improvement Company; Mr. A. M. Mattice, late of the Westinghouse and of the Allis-Chalmers Machine Companies; Mr. W. M. McFarland, late of the Westinghouse, and now with the Babcock and Wilcox Company, and Mr. A. M. Hunt, of the Naval Consulting Board.

The older engineers of the Navy did their share in the work of development, but they were growing old and disappearing rapidly from the active field, so that the brunt of the full development of our naval machinery as it exists today fell upon the shoulders of these few Naval Academy classes who have been in the battle from its inception. How well they have carried out their task is testified to by the efficiency now being demonstrated by our vessels of all types actively engaged in actual war service.

NAVAL ENGINEERING SUBSEQUENT TO THE CIVIL WAR.

In 1863 the Navy Department had designed and had laid down several cruisers whose maintained sea speed was specified to be higher than that of any vessels, naval or merchant, afloat. The first of these vessels was not completed until 1868. This was the *Wampanoag*, the most celebrated in marine engineering history on account of the phenomenal results obtained, phenomenal in those days as compared with existing vessels. The machinery consisted of two single-cylinder simple engines, of 100 inches diameter of cylinder and 4 feet stroke, direct acting, and connected to the main propelling shaft by multiplying gearing, the designed revolutions of the engines being 30, piston speed 240 feet per minute, revolutions of the propeller shaft 60. The gears were made with wooden teeth in order to deaden the noise.

On trial the results obtained were as follows:

Duration of trial	37½ consecutive hours.		
Wind and sea	Fresh wind abeam, moderate sea on quarter.		
Average speed, knots	16.71	....	11.39
Average revolutions, engine	31.06	....	21.36
Average revolutions, propeller	63.673	....	43.78
Boiler pressure, pounds per square inch	31.97	....	20.69
Coal per hour, pounds	12,690	....	3,474
Coal per day, tons	136	....	37.2
Knots per ton of coal	2.95	....	7.34+
I.H.P.	4,048		

What these results mean will be more fully appreciated when it is considered that this maximum speed was not attained in the British Naval Service until eleven years later, 1879, with the large despatch vessels *Iris* and *Mercury;* in the merchant service in 1879, with the Guion liner *Arizona;* while in our own Naval Service a period of 21 years elapsed before this speed was again equalled.

The *Wampanoag* and her sister ships were born too soon. They were surveyed and condemned as unfit for the Naval Service within a year of their trials, among the reasons given

for this action being "excessive amount of fuel burned" and "the grate surface exceeded the area of the midship section of the ship." Viewing the fuel consumption in the light of today, the first reason appears childish, but it must be remembered that, at the time of these vessels, the leading spirits of the Navy were the developments of the days when sails as a motive power were supreme, and machinery was only looked on as an auxiliary to help out in time of calm or when the use of sails was impractical. The second reason appears peculiar today, but may have had some connection with the subject according to the viewpoint of those days of *long ago*.

The machinery of the *Wampanoag* was designed by the late Commodore B. F. Isherwood, U. S. Navy, who at that time was Engineer-in-Chief of the Navy. He was subjected to a great amount of criticism at the time but, as we know today, he was ahead of his time and his accomplishment was not appreciated at its full value.

The first decided and permanent step in advance in marine engineering in our Navy occurred in 1871–1874, when it was determined to throw away the two simple cylinders of each of six sets of engines which had been built for installation in six old sloops being rebuilt under the head of repairs, and to fit in place of them a high and a low-pressure cylinder. This was the first appearance in our Service of the compound engine.

These engines were of what was known as the "Isherwood" type. They were horizontal, back acting, with the cross-tail guides located under heavy rectangular cast-iron condensers on the opposite side of the ship from the cylinders. The main circulating, main air and main feed pumps were all of the reciprocating type, worked directly by rods taking hold of lugs on the side rods connecting crosshead and cross tail. The stroke of the engines was 42 inches, the revolutions about 50, giving a piston speed of 350 feet per minute. The steam pressure at the boilers was 80 pounds per gage. This rise in pressure from the 30 or 40 pounds carried with the old simple engines necessitated a departure from the rectangular type of

boiler which had been used in the past, and the adoption of boilers of the cylindrical type. These cylindrical boilers were classed as "Compound" boilers, but were the same as the well-known "Scotch" boiler of today.

In 1873 Congress passed an act authorizing the construction of eight vessels of war. These were the vessels referred to earlier in this lecture as coming into the Service in 1876-77. The most noteworthy thing in this act of Congress was the fact that the act required that these vessels were to be "steam vessels of war, with auxiliary sail power." This proviso indicates the development of viewpoint that was occurring and is one of the first signs of a full appreciation by Congress and by the naval authorities of the value of the steam engine as a main power unit for marine propulsion.

The three steel sloops were fitted with engines of 560 I.H.P., those of the four wooden ones with 800 I.H.P., while the wooden frigate was given 3,000 I.H.P. All of the engines were of the then well-known Isherwood type, as already described, the sloops being two-cylinder compound while the steam frigate was three-cylinder compound, all of the engines being horizontal.

Up to this time and for several years later all engines for our fighting ships were made horizontal in order to keep them below the water line of the vessel and thus insure a certain degree of protection against gun fire. The engineer departments of the ships were of the very simplest type, practically all principal pumps being operated from the main engines. The auxiliary machinery was very limited in amount, there being no such machines or apparatus as blowers, dynamos, evaporators, ice machines or even sanitary pumps. Distilling condensers were fitted, but the steam for them was taken directly from the main boilers and the fresh water thus made was used only for officers bathing, for cooking, and drinking purposes.

Simple as were these machinery installations, I still have a vivid picture in my mind of the engineers of those days walk-

ing around thoroughly bowed down by the weight of responsibility carried on their shoulders. A Junior Officer of today would feel very much injured should he be ordered to duty as Chief Engineer of one of these old vessels, some of which are still in active service, and wonder as to what was the trouble with his record.

In June, 1883, the Navy, while not dead, was moribund, but in that year occurred what may be styled

THE RENAISSANCE OF THE NAVY AND OF NAVAL MARINE ENGINEERING.

By 1883 the condition of the Navy had become so bad that officers were ashamed to take their vessels into ports where the vessels of foreign nations might be encountered, so antiquated were our ships, not only in appearance but in material of all description.

During this year, however, the interest of Congress awakened from its long sleep of nearly twenty years, and the naval appropriation bill of that year carried in it a provision for the construction of four new vessels. These ships when built were known as the *Chicago, Atlanta, Boston* and *Dolphin;* the first three being protected cruisers and the fourth a despatch vessel.

The *Dolphin, Boston* and *Atlanta* were all single-screw ships, the first named having a 2-cylinder, vertical, inverted compound engine of 2,300 I.H.P. at 75 revolutions and 110 pounds steam pressure at the engines. Those of the *Atlanta* and *Boston* were similar in design to those of the wooden steam frigate of 1876, developing 3,500 I.H.P. at 70 revolutions, with 90 pounds pressure at the engines, the piston speeds being 600 feet per minute for the *Dolphin* and approximately 500 feet for the two cruisers.

The questions of the design of the hulls and machinery of these four vessels which formed the nucleus of our new Navy, were of such importance that in the appropriation bill a provision was made for a Naval Advisory Board to determine these important questions. As organized, this Board con-

sisted of five Naval Officers, three being of the Line, one Engineer and one Naval Constructor, associated with two Civilian members, one of these being a Naval Architect while the other was a Marine Engineer, both being very prominent in their professions.

As can be seen, the Board was very conservative in the cases of the *Dolphin*, *Boston* and *Atlanta*, but in the case of the fourth vessel, the *Chicago*, they allowed themselves to go to the other extreme in their desire to show something peculiarly *American*, and widely different from anything in the line of machinery that had heretofore been fitted in the vessels of any Navy. Anxious as they were to show progress, the engines which they adopted were remarkably like the engines which were fitted to a craft known as the *Stevens Battery*, thirty years prior to this time.

The civilian marine engineer was given practically full control in the determination of the type of machinery, due, as was stated by a member of the Board, "to the popular pressure to give the outside a chance."

The vessel was twin screw, with two two-cylinder compound engines, one to each shaft. The cylinders stood vertically on the bottom of the ship, piston rods passing through the upper heads and connecting by means of links to athwartship walking beams which connected through links to the crank shaft on the opposite side of the vessel. The engines had a stroke of 57 inches and were designed for about 70 revolutions when developing 2,500 I.H.P. each, the piston speed being 665 feet per minute with a steam pressure at the engines of 90 pounds per gage.

The boilers were also of a peculiar type, there being five double-ended and two single-ended ones. They were cylindrical, return tube, with internal back connections. The boilers were set in brickwork and externally fired. There was one point of real advance made, however, and that was in fitting the vessel with forced draft, which had been installed in isolated cases in the past.

The life of the entire installation was short, it being entirely removed after seven years' service and replaced with inclined three-cylinder triple-expansion engines, and a mixed installation of Scotch and of water-tube boilers, the latter being of the early Babcock and Wilcox type.

The supporters of sail power still being in the ascendant, although the signs of the times indicated their speedy downfall, these four vessels were all rigged powerfully, the *Chicago* being a bark, the *Atlanta* and *Boston* brigs, and the *Dolphin* a topsail schooner.

The next vessels appropriated for, in 1885, the *Newark*, *Charleston*, *Yorktown* and *Petrel*, marked a distinct era in the history of naval marine engineering in this country. The *Charleston* and *Petrel* were the last fighting ships designed with compound engines, while the *Newark* and *Yorktown* were the first vessels to be fitted with engines of the triple-expansion type. The engines still were horizontal.

The steam pressure carried in the boilers of the compound engined vessels was 90 while in those having triple-expansion engines it had risen to 160. The piston speed of the *Charleston* was 690 feet per minute while that of the *Newark* was 826. The power of the *Charleston* was designed 7,500, that of the *Newark*, 8,500. Both of these vessels on trial exceeded the record Navy speed of the old *Wampanoag*, the first in our Navy to do this after an interval of 21 years.

In 1883 electric apparatus for ship's lighting had been installed on the *Trenton*, the steam frigate of 1876, and she had the honor of being the first sea-going vessel, naval or merchant, to be so equipped. The *Yorktown* also marked the introduction of a new auxiliary. On her was installed the first evaporating set. In the vessels prior to the *Yorktown* and in those contemporaneous with her, the steam for the distilling condensers had been taken from the main boilers or from auxiliary boilers provided for this purpose and for general auxiliary work. On the *Yorktown*, however, special evaporating apparatus as it is known today was installed, and such

apparatus was gradually installed on all vessels of any size in the Naval Service.

As the *Charleston* and *Petrel* marked the passing of the compound engine from general use in the Navy, so the *Newark* and *Yorktown* marked the coming of a new type, which lasted for many years. They may be said to have been the first of the

"ERA OF THE MULTIPLE-EXPANSION RECIPROCATING STEAM ENGINE."

The engines of these two vessels were of the same type, three-cylinder, horizontal, triple-expansion. The *Newark* and *Yorktown*, copying the *Charleston*, which had introduced to us a new type of valve gear, were fitted with a radial type of valve gear known as the Marshall. This operated with one eccentric and one eccentric rod and had the one good feature of maintaining constant steam and exhaust leads at all degrees of linking up. It had, however, the objection of a great number of joints which, after adjustment, required a careful resetting of the valve. The popularity of this gear was short lived and it soon disappeared from our designs.

In looking for the machinery designs for these vessels the Navy Department, not having been impressed with the designs carried out in the first lot of new vessels, went abroad for the designs for the machinery for the *Charleston*, and the designs were obtained from England. These were supposed to be the designs used for the Japanese cruiser *Naniwa-Kan*, but they were in reality made up as a composite set taken from four different vessels, the Italian cruisers *Etna* and *Giovanni Bausan*, the *Naniwa-Kan* and a fourth ship, supposed to be the Chilean cruiser *Esmeralda*. Upon attempting to fit this hodge-podge collection together numerous cases of interference occurred, but finally a satisfactory vessel was evolved, although before this was accomplished the original air pumps, which were of the horizontal, high-speed type, were discarded and air pumps of the vertical, inverted, overhead-flywheel type substituted.

The machinery of the *Petrel* was designed by the Navy Department while those of the *Newark* and *Yorktown* were the children of the brain and skill of Mr. Horace See, at that time the Designing Engineer for the Cramps Ship & Engine Building Company, the builders of these two ships, and to Mr. See belongs the credit for fathering the multi-stage expansion era in the Navy.

The year 1886 was a banner year for the Navy. The Naval appropriation bill for that year provided for one second-class battleship, one armored cruiser, one protected cruiser, one dynamite-gun cruiser and one torpedo boat. Of these vessels four were practically new types of vessels in our Service, for the nearest approach to battleships we possessed were the large monitors already mentioned, of which at this time four were being completed; no such craft as the armored cruiser had existence with us. While we had had torpedo boats previous to this time, they had usually been either of the make-shift variety or of some disappointing freak type from which nothing could be expected. The dynamite-gun cruiser may be placed in the freak class. She was "wished" on us and was a failure, so far as her special function was concerned, from the outset.

The engines of the armored cruiser *Maine* and the dynamite vessel *Vesuvius* were designed by the Bureau of Steam Engineering of the Navy Department and the Cramps Ship and Engine Building Co., respectively, those of the torpedo boat *Cushing* by the Herreshoff Manufacturing Company, while those of the battleship *Texas* and of the protected cruiser *Baltimore* were purchased in England.

The engines of the *Maine* and of the *Texas* were forerunners of the passing of the horizontal engine for main propelling purposes on war vessels of our Navy. These engines were vertical, inverted, three-cylinder, triple-expansion engines, twin screw, having a piston speed of 800 feet per minute for the *Texas* with a steam pressure of 150 pounds at the engines. The piston speed and steam pressure of the *Maine* were 750

feet and 135 pounds. The *Cushing's* engines were twin-screw, five-cylinder, vertical, inverted, quadruple-expansion engines, a new departure for us, and twin screw; the boilers were of the old style dry-tube Thornycroft type, the first appearance in the American Navy of the small-tube express type of boiler for main power units in other than launches. The *Cushing* marked a still further increase in piston speed and in steam pressure, the first having risen to 928 feet per minute, and the latter to 250 pounds per gage.

Of the vessels ordered during the next year, 1887, the cruisers showed a great decrease in sail power over the *Newark*, which had been bark rigged. The only advance in engineering that was made was in the case of a monitor, the *Monterey*, in which, for the first time in one of our comparatively high-powered vessels of any great fighting value a large proportion of the power was supplied by water-tube boilers. On this vessel two-thirds of the total power was supplied by water-tube boilers manufactured by the firm of Charles Ward, of Charleston, West Virginia. They were of the cylindrical multi-coil type, and while giving satisfactory results until 1903, when they were replaced with water-tube boilers of a later type, they possessed several objectionable features,—they were difficult to keep clean, both exterior and interior of tubes; tube renewal was difficult; on account of cylindrical form, the floor space necessary for a given amount of grate was excessive. While these boilers were successful, sea-going men, both deck and engineers, are very conservative, and it was several years before the water-tube boiler succeeded in displacing the Scotch boiler in naval vessels, and it is now struggling very hard to take the place of its competitor in the merchant service.

In 1890 was made the first attempt to attain maximum economy in propulsion at low speeds by means of dividing the total power on each propeller shaft into two units. This was done in the case of the *New York*, now the *Rochester*. This

vessel was the first of our armored cruisers of the true type, and was followed in 1893, by the *Brooklyn*, having a similar arrangement of engines. There were four vertical, inverted, three-cylinder, triple-expansion engines, two to each shaft, with a shifting coupling between the forward and after engines in order that at low powers and speeds the forward engines could be disconnected and only the after engines used.

We again find an increase in steam pressure and piston speed over what had been previously used, the former having risen to 175 pounds, while the latter had reached 950 feet. The total power had risen to 16,000 I.H.P.

The arrangement of machinery, while very satisfactory for peace times, failed under the test of war, as it was not always possible, when the emergency call for high power and high speed was heard, to take the necessary time to couple up the dead engines, which also must be kept ready warmed up for just such an emergency.

The *New York* marked the introduction of a new and very successful type of air pump into the Navy, and a type which has continued until the present day. The legend is that the idea for this type came from Passed Assistant Engineer Stacy Potts, U. S. Navy, who was on duty in the Bureau of Steam Engineering at the time the plans of machinery for the *New York* were being laid down. The idea, however, was worked out to successful completion by the Blake Pump Company. These pumps were the first of the double, vertical, inverted steam-cylinder type with a rocking beam connection between the two pumps forming a unit. This method of connection rendered the work being done per double stroke much more constant and resulted in a very much smoother running pump than those used in the past.

This same year witnessed the entry of another new auxiliary into the Service, one that, at the time, was regarded as a luxury, but one which the experience of years has demonstrated to be an absolute necessity in solving the problem of maintain-

ing contentment in a ship's crew. This new auxiliary was the "refrigerating machine." The unit fitted was a small one, being what is known as the "Allen Dense Air" machine of one ton capacity. It was fitted on board the *Chicago*, and the results obtained were so satisfactory that its use in gradually increasing capacities was continued until the field became so attractive that the CO_2 machine manufacturers finally entered it. Both types, "Dense Air" and "CO_2," are now used on our larger vessels, while on vessels of the destroyer type and all other vessels of very small refrigerating requirements, small SO_2, ethyl chloride and CO_2 machines are installed.

No changes of importance, with one exception, occurred in the machinery installations of our vessels until 1896, when a newcomer in the marine boiler field was installed on board the *Chicago* in place of the externally-fired cylindrical boilers originally fitted. The original engines were taken out at the same time and were replaced by inclined three-cylinder, triple-expansion engines.

The new arrival in the marine boiler field was the Babcock and Wilcox boiler. As installed in the *Chicago*, this boiler differed considerably from the Babcock and Wilcox boiler as we are familiar with it today. These original boilers had the steam drum located at the back of the boiler at the top; the headers were sinuous, but were arranged vertically with the tube faces stepped in order that the inclined tubes would enter the tube sheets perpendicular to the plate face; the tubes inclined upward from back to front; and in order to maintain sufficient distance between tubes and grates at the back, the grates were heavily inclined downward towards the back of the boiler. The sides of the furnaces were formed by steel water boxes, square in section. A feed-water heater, built into the boiler casing and located in the uptake passage, was also fitted. There was no baffling fitted among the tubes, but later, table baffling, that is, baffling lying on the tubes, was installed.

The exception in the matter of improvements was the introduction of feed-water heaters. These were first called for in the machinery specifications of the *Oregon* class of battleships, but were not fitted. The first installation made was on the *Iowa* and was such that it would not be even considered today. The heaters were of the single-pass type, that is, the water flowed through the tubes in one direction only. The heating medium was auxiliary exhaust steam. The method of installing was as follows: A pump fitted for both fire-and-bilge and for auxiliary feed purposes had a suction connection to the main feed tank and a discharge to the auxiliary feed line with a branch to the feed heater. The water, after being heated, passed back again and mixed with the unheated water in the main feed tank, from which the main feed pumps took their suction.

This arrangement was faulty in the following details:

1. It required the operating of two pumps in place of one to heat the water and to deliver it to the boilers.
2. The radiation losses of heat were increased.
3. It prohibited the possibility of high feed temperatures being attained.
4. The double function of the auxiliary feed pump, that is, both fire-and-bilge and feed, should never be allowed, due to the danger of salting up the boilers through leaky valves.

This arrangement of feed heater was not adhered to for any length of time, being soon superseded by the following arrangement: A hot-well pump was installed between the main feed tank and the feed heater, pumping the water through the heater directly to the main feed pump, and thence to the boilers. This arrangement also was objectionable on account of requiring an additional pump, although it kept down the strains on the feed heater.

In still later ships the feed heater was installed in the main feed line between the main feed pump and the boiler, and this is its location at the present time.

During this same period, 1890 to 1896, were built the two fast cruisers *Columbia* and *Minneapolis*, noted as the first vessels of the modern Fleet to have the propelling power divided among three shafts. The idea of this sub-division was to give greater elasticity in division of power for cruising in order that greater economy at these speeds could be realized; thus, at low speeds only the center engine would be used, the two wing screws being uncoupled and allowed to turn idly. For middle speeds the two wing engines would be used, the center screw turning idly, while, of course, for full power all engines would be in use. These vessels made considerably over 22 knots on trial and marked a decided increase in this respect over any large vessels we had previously built. Whether the expected economies were realized is a matter of question.

In 1894, among other vessels appropriated for, was the gunboat *Nashville*. The machinery designed for this vessel marked an attempt at improvement which was doomed not to be repeated. The engines were vertical, inverted, quadruple-expansion, so fitted that the low-pressure engine could be disconnected and the remaining three cylinders used as a triple-expansion engine at low powers. The steam pressure was increased to 250 pounds. The changing type feature, like all such arrangements, might just as well have been omitted, as it was seldom, if ever, used, and in 1907, when the ship was reboilered, the independent exhausts from the two I.P. cylinders to the condenser were eliminated and the engines made fixed quadruples.

Until 1896 all large engines of the vertical type in the Naval Service had been built either with cast or built-up plate columns front and back or with this type of column at the back and forged columns at the front of the engine. In 1896 five battleships were appropriated for. On two of these, the *Kearsarge* and *Kentucky*, were installed engines which were the first to be built having all columns of the engine framing forged and

turned, with cross-tie bracing between, instead of following previous practice as above outlined. This new style of framing had previously been used for small engines up to and including those for gunboats. The further development to battleship size was made by Passed Assistant Engineer (now Rear Admiral) F. H. Bailey, U. S. N., and although severely criticised by many engineers at the time on the ground of probable lack of rigidity, has been found so satisfactory that it has been adhered to for all later construction, the engines of the *New York* and *Texas*, each engine of 14,000 designed I.H.P., being the largest built.

The *Kearsarge* and *Kentucky* are of interest in other than engineering matters as, so far as battery arrangement is concerned, they were the forerunners of our present Dreadnaughts. While their main battery was of mixed caliber of gun, 12 inches and 8 inches, it was all arranged on the center lines of the vessels, the 8-inch turrets being carried on the top of the 12-inch ones. Previous battleships had had the smaller caliber guns of the main battery carried in turrets located at the sides of the ships, usually at the corners of the central citadel in which were carried the lower-caliber guns forming the secondary battery.

The five vessels together are of interest as they mark the definite period in the Naval Service when the use of Scotch boilers for production of steam for high-powered units was abandoned. No fighting ships subsequent to these five had any other than water-tube boilers. The policy as to type of water-tube boiler to be used for the different types of vessels vacillated considerably for a time, but has finally settled down to a few thoroughly well tried and satisfactory ones.

APPEARANCE OF THE SUBMARINE.

In 1895 the Navy Department, for the first time since the Civil War, began to take an interest in the submarine boat. An inventor, Mr. John P. Holland, had been experimenting for several years previous to this time with the problems of the

submarine, and had formed a company known as the "John P. Holland Submarine Boat Company," for their construction. The Navy Department, in 1895, entered into contract with this Company for the construction of one submarine boat having the following characteristics:

Length, ...84 feet;
Diameter, ...11 feet 6 inches;

the section being circular.

The speed on the surface was to be 15 knots, awash 14 knots, and submerged 8 knots for six hours continuous running.

The surface propelling was to be accomplished by means of steam engines, of which there were three, the two wing engines driving the wing screws being of 600 I.H.P. each, while the center engine, driving the middle screw and also to drive the dynamo for charging the storage batteries, was of 300 I.H.P. All engines were of the vertical, inverted, three-cylinder, triple-expansion type.

The submerged propulsion was provided by electric motor deriving its current from the storage batteries.

The steam was provided by one Mosher water-tube boiler having 3,000 square feet of heating surface.

Among other requirements, the Navy Department specified as follows:

1. Sufficient structural strength, with a reasonable factor of safety, in order to withstand hydrostatic pressure at a depth of immersion of 75 feet.

2. Ample stability in both the surface and submerged condition.

3. Automatic arrangements for preventing submergence beyond a predetermined depth.

4. Automatic compensation for stores or ammunition consumed, and independent mechanism for quickly correcting any variations in trim due to shifting of weights or other causes.

5. Efficient mechanical means for steering a fixed course.

6. Ample provision for the supply of fresh air to the crew—this to be accomplished chemically, or by means of air stowed under pressure in tanks.

7. Efficient apparatus, independent of air pumps or air pressure, for checking submergence and causing vessel to rise rapidly to the surface.

8. Ability to maintain an approximately fixed position, also a definite depth of submergence without undue expenditure of power.

9. Provision for escape of crew in the event of serious damage to hull or derangement of machinery.

This child of our dreams was still-born. The submarine was not yet due to arrive nor did it arrive until the gasoline engine of sufficient power for propelling purposes was perfected, an event which occurred several years later.

THE SPANISH WAR.

Before any further progress was made in engineering in the Navy that Service was subjected to the short but strenuous ordeal of the war with Spain.

While the engineering material of the time stood up well under the strain of practically continuous operation under the necessity of each unit being self-supporting, yet the war brought out some faults which were carefully avoided in later construction, faults not only in machinery but also in types of vessels.

1. For instance, the building of such types of vessels as the dynamite cruiser *Vesuvius* and the armored ram *Katahdin* was shown to be a useless waste of money, these two vessels demonstrating that they had no military value whatsoever.

2. The value of the monitor as a sea-going fighting craft was exploded. These vessels while of value for fighting purposes in smooth, still water of harbors, roll so quickly in a sea way that it is practically impossible to keep their guns bearing on any given target, while, due to their low free board, they are particularly hard on the crews which may be unlucky

enough to be detailed to man them. In addition, vessels of the monitor type must be limited in speed, because to attempt to drive them at high speed would practically make them submarines so far as their crews were concerned.

This exposure, by actual sea service war conditions, of the monitor fallacy, was a blessing to the true friends of the Navy, as, up to that time, the advocates of the battleship had been forced to fight with vigor to retain battleship provisions in the naval bills. The advocates of the monitor were always attempting to replace the battleships by monitors on account of the much less cost of the latter type. They never could forget that the original little turreted *Monitor* had more than held her own against the make-shift casemated ship *Virginia*, ordinarily known by her original name as the *Merrimac*, a vessel having such faulty machinery and of such small power that she could hardly get out of her own way.

FAULTS IN MACHINERY DEVELOPED BY THE SPANISH WAR.

As stated, the war demonstrated the existence of some faults in our machinery installations, among the most glaring of which may be mentioned—

1. The use of Scotch boilers on war vessels when boilers of types which will permit of rapid raising of steam from dead boilers and high degree of forcing are at hand.

2. The absolute necessity of large evaporating and distilling plants to insure an ample supply of fresh water.

3. The desirability of large refrigerating and cold-storage capacity on board fighting ships in order to preserve the health and contentment of the crew.

4. The fallacy of providing such arrangements of power units as existed on the *New York* and *Brooklyn*.

Faults 1 and 2 were of serious consequences to one of our battleships, 1, 2 and 4 to one of our armored cruisers, and fault 4 to the other armored cruiser at the battle of Santiago when we destroyed the Spanish Squadron.

Due to the lack of sufficient evaporating and distilling plants on our large vessels it had become necessary during the war to make direct connections between the auxiliary steam mains and the main and auxiliary condensers of the vessels in order that live steam could be taken from the boilers and condensed by these condensers, making up the boiler water thus used, by salt feed taken directly from the sea. This prostitution of the main boilers resulted in the formation of heavy scale on the water sides of their heating surfaces and necessitated frequent emptyings and openings of the boilers in order that as much of this scale as possible might be removed.

When on July 3, 1898, the Spanish ships emerged from Santiago harbor, the battleship in question was lying with low fires under about one-half of her boilers, the other boilers being cold and some of them being opened up for a necessary scaling. The first armored cruiser was in just as evil a plight, so far as her boilers were concerned, but she was still further handicapped by having the forward engine on each side disconnected from the shaft. The second armored cruiser had proceeded several miles to the eastward of Santiago harbor, having fires in good condition under half her boilers while the other boilers were dead and cold. She also had her forward engines uncoupled.

Immediately upon sighting the outcoming vessels, the battleship and first armored cruiser dashed for them with what speed they could command, the Engineers' forces tumbling to their stations, starting forced-draft blowers, closing firerooms, closing up and filling boilers, laying and starting fires. The case was too urgent, however, for the armored cruiser to permit of her stopping for sufficient time to couple up her forward engines.

The pace was too rapid for the battleship. Before she could even build up the fires she had going her speed had fallen to about seven knots, she had hopelessly dropped behind and was out of the fight for good.

The first armored cruiser was apparently in somewhat better condition. Her fires were evidently in better shape and answered more readily to the forced-draft call. Having one-half of her engines and one-half of her boilers in use the balance between engines and boilers was satisfactory. She held her position in the chase and added on boiler after boiler as soon as they equalized their steam pressures with that in the main steam lines. It was all to no purpose, however, as the boilers originally in use were giving all the steam the active engines could take. The result was that this magnificent cruiser of 22 knots, under normal conditions, found herself humiliated by being strained to the utmost to hold herself up even in the chase with the battleship *Oregon*, whose trial speed had been approximately 15 knots, but whose speed on this day, due to loading of the ship, foulness of bottom and propellers only approximated 14 knots.

The departure of the second armored cruiser coupled with the absence of the *Massachusetts*, which vessel was at Guantanamo Bay coaling, had so weakened our blockading line that the Spanish Admiral was unable to resist the temptation to try his long contemplated dash for Havana and the destruction of our weak blockading force on the north coast of Cuba.

Immediately the second armored cruiser discovered the van of the Spanish ships coming out she swung around and headed for the westward. Realizing that she was severely handicapped by her initial distance from the scene of operations and that she would need every atom of speed at her command to put her into the fight as soon as possible, and remembering the old adage that "*a stern chase is a long one*," she stopped after turning, coupled up her forward engines, losing valuable minutes in the operation, lighted fires under her dead boilers and headed for the scene of action with what speed she had available. As steam formed and rose to an equality with that in the main steam lines, boiler after boiler was cut in, the engine throttles being still further opened to take advantage of the

increased steam supply. The speed increased with every additional boiler added until at the time the last Spanish ship hauled down her flag and headed for the beach, this cruiser had risen from about 13 knots to a speed of 18 or better and showed every evidence of still improving.

Had the *Colon* done that which it was in her to do this second armored cruiser would have been the only obstacle between her and the destruction of our northern blockading squadron, as she outgunned and outspeeded every ship we had in that squadron with the exception of the monitors, and those she could easily have avoided.

Satisfactory boilers for use on fighting ships must possess the following characteristics:

1. Maximum power on minimum weight.
2. Maximum power on minimum floor space.
3. Ability to stand high forcing without undue straining.
4. Assured positive circulation of contained water.
5. Sufficient contained water to permit of short stoppages of feed supply.
6. Sufficient steam space to insure against priming.
7. Ease of repairs.
8. Easily cleaned on both fire and water sides.
9. Renewal or installation of units without dismantling and opening up of vessel.
10. Sufficient elasticity in form to permit of raising steam from cold water in a short time without producing leaks.
11. Such form as will reduce danger to vessel from broaching of a boiler under steam, to a minimum.

To any one familiar with the Scotch boiler it will be at once apparent that this type does not meet the above requirements except in two points, those numbered 5 and 6, and in those the type provides very much in excess over what is actually necessary. The water-tube boiler, on the other hand, when properly designed, measures up to every requirement, so that when judgment was finally held on the two types, it was a foregone conclusion that the Scotch boiler would be the loser.

Such was the case immediately subsequent to the Spanish War, and the Scotch boiler disappeared from all our construction with the exception of auxiliary vessels, and in the past few years it has even disappeared from these.

ADOPTION OF WATER-TUBE BOILERS FOR CAPITAL SHIPS.

Shortly after the beginning of the Spanish War Congress passed a bill appropriating for the construction of three battleships, the plans and specifications for which were prepared by the firms which later received the contracts for the construction of the vessels, and which plans were submitted to and approved by the Navy Department.

These vessels were the *Maine*, *Missouri* and *Ohio*, the first named being designed and built by Cramps Ship and Engine Building Company, the second by the Newport News Shipbuilding and Dry Dock Company, and the third by the Union Iron Works. All three were fitted with water-tube boilers, those of the *Maine* being of the Niclausse type, which were authorized notwithstanding vigorous protests made by the then Engineer-in-Chief of the Navy, the late Rear Admiral George W. Melville, while those of the *Missouri* and *Ohio* were of the Thornycroft type with tube steam ends of the outer half of each nest of tubes leading into the steam drum well above the water level. The forced draft on the *Maine* was of the closed stoke-hold arrangement, while that used on the other two vessels was the Howden Hot-Air System, the fans taking their supply from the stoke-holds and discharging it through heater boxes located in the uptakes, to the ash pits of the boilers.

The boilers with other fireroom arrangements on the *Missouri* and *Ohio* were very satisfactory and are as satisfactory today as when given their first trials.

The *Maine's* history is another story. Her boilers were unsatisfactory from the first, and several years ago were removed and boilers of a tried type installed in their place. In fact every Niclausse boiler originally installed in the ships of our Navy has been removed and this tried type of boiler installed.

The structural feature in the *Maine's* boilers which rendered them so unsatisfactory in operation was the design of the front headers. These were of cast steel, and were of insufficient area to accommodate the necessary volume of down coming current of water and up rising circulation current of water and steam having such slow velocity of flow as exists in boilers of the free circulation type. The result of this restriction of circulation areas was the constant bending, burning and bursting of generating tubes which, of course, in turn affected the *morale* of the engineer's force and the general efficiency of the vessel. In later vessels fitted with this type of boiler, of which there were two battleships, two armored cruisers and one monitor, the design of the headers was altered to provide freer circulation and the material was changed to forged steel. These boilers were much more satisfactory than the *Maine's*, but all of the Niclausse type are unsatisfactory under forced-draft conditions on account of the inefficient baffling which reduces their economy under such conditions. Boilers of the Niclausse type and all types of boilers using what is known as the Field tube, have departed from our Service and it is extremely doubtful whether we shall ever revive their use.

The engines fitted to two of these vessels each represented a particular type of the vertical, inverted, triple-expansion engine as it had existed in the Service up to and including that time. The engines of one of the vessels, so far as the arrangement of cylinders was concerned, was a new comer.

The engines of the *Maine* were three-cylinder engines with the high-pressure cylinder forward, then the I.P. and farthest aft the L.P., with the sequence of cranks according to this same order. The back columns of the engine were inverted Y, cast-steel columns with turned forged front columns. All of the valves were single-ported piston type.

The engines of the *Ohio* in their arrangement of cylinders were similar to all vertical, four-cylinder engines which we had

built up to that date, the number being very small. The engines were four-cylinder, vertical, inverted, triple-expansion with the H.P. cylinder forward, then the I.P. followed by the forward L.P. and that in turn by the after L.P. The H.P. and I.P. cranks were opposite each other while the two L.Ps. were opposite each other and at 90 degrees to the H.P. and I.P. The framing was of a type similar to that of the *Maine's* engines. The H.P. valve was of the single-ported piston type, while those of the I.P. and L.P. cylinders were of the double-ported flat slide-valve type with balance rings on the backs.

Both types of engines are defective in one particular, and that is the balance, the *Ohio* type being worse in this particular than that of the *Maine*. With the *Ohio* type we have really the equivalent of a two-crank engine with the cranks at right angles and very unequally loaded; unequally loaded, that is, when the division of power among the four cylinders is such as is usually sought with a triple-expansion engine.

With the *Missouri* arrived a type of engine which, so far as general arrangement, was to endure with us until the reciprocating engine, just as it had reached its maximum attained efficiency, was forced from the field of usefulness by abnormal increases in power required per unit, and by the superior combined efficiencies of propellers and engines of other types.

The engines of the *Missouri* were designed by Chief Engineer C. F. Bailey, of the Newport News Shipbuilding and Dry Dock Company, and were of the four-cylinder, vertical, inverted, triple-expansion type, as were those of the *Ohio*, but the cylinder arrangement of the *Missouri* was as follows: Beginning forward the order was forward L.P., H.P., I.P. and after L.P., the cranks being arranged at 90 degrees, with the forward L.P. and H.P. opposite each other, and the I.P. and after L.P. opposite each other. The valves were all of the single-ported piston type, operated by Stephenson valve gear. The framing was similar in construction to that of the *Kearsarge* and *Kentucky*, being formed of turned forged-steel col-

umns, front and back, tied together by the necessary steel tie rods, and with the crosshead guides carried on strong backs securely fastened to the back columns.

THE DISAPPEARANCE OF SAIL POWER.

With the passing of the Spanish War passed forever the fitting of sail power to any of our fighting ships with the exception of slow-speed and low-powered gunboats.

With the passing of sail power passed away crews of the character with which we older men had long been familiar. No more on our decks was seen the hoary headed "old shell back." The lusty topmen and agile light-yardman had disappeared, their places being usurped by the competent turret captain, the keen eyed, alert, gun pointer and sight setter, the machinist, water tender, fireman and his satellite the coal passer. The "Black gang" had come into its own. The old Engineer Corps disappeared, being absorbed into the Line. The central spring of ship life was the boilers, her life blood was steam. Every officer and man on board became more or less an engineer, the very nature of practically every duty compelling them to do so whether they wished it or not.

From the end of the Spanish War until the present day the advance in Marine Engineering has been phenomenal. At the first the changes were few, consisting mainly of a gradual increase in power for Capital Ships, from the 15,000 I.H.P. of the last three vessels mentioned up to 19,000 I.H.P. for the next class, the *Georgia* and sisters, and still higher to 25,000 I.H.P. in the cases of the armored cruisers. In the same period, between 1899 and 1905, the boiler pressures increased to 295 pounds, and the piston speeds rose to 960 feet per minute for large engines, to 1,000 feet for torpedo boats and to 1,200 feet for destroyers.

Of the two latter classes of vessels, 18 small torpedo boats had been built up to and during the war, but had been of no special value during that period, except the use made of the

larger ones as despatch boats. During the war appropriation was made for 15 more of this type, but of higher speed and larger size and power.

Some of the earlier boats had been very successful, so far as results in speed and machinery were concerned, and these were the vessels built by the Bath Iron Works under the careful supervision and from the designs prepared by Mr. Charles P. Wetherbee, then, as now, Chief Engineer of that Company, also the vessels built by the Herreshoff Manufacturing Company, of Providence, R. I.

The same appropriation bill carried in it a provision for 16 Destroyers, the first of this type of vessel to be authorized for our Navy.

The characteristics of the vessels were about as follows:

	Length, feet.	Displacement, tons.	I.H.P.	Speed, knots.
Torpedo boats	157–225	165–340	3,000–7,200	26–30
Destroyers	240–248	408–433	7,200–8,400	28–29

The engines of all were of the three-cylinder, vertical, inverted, triple-expansion type with attached air pumps worked directly from the main engines in the cases of all the destroyers and some of the torpedo boats while in others of the torpedo boats they were driven from the main shafts.

According to the standards of the period, some of these vessels were rated as very successful, but all of them had, to a greater or less degree, all of the troubles which are inherent in high-speed reciprocating engines of such light construction as the machinery of these types of vessels must be. If any of them came through a four-hour full-speed trial without anything of greater importance than the complete breaking down of a forced-draft blower engine it was considered a matter deserving of the greatest commendation. Such successful trials were very few and far between.

Those were the days before boilers of the water-tube class had settled down into a few generally accepted standard types,

and when every engineer was anxious to design his own boiler. As a result these vessels were equipped with several different types, such as the old style Thornycroft, commonly known at that time as the "Daring," the Normand, a mushroom variety of Mosher, the Seabury and the Fore River. Of these boilers we find in common use today as Standards, only two, the Normand and the Thornycroft, but the modern form of the latter boiler differs widely from that of 1900.

A few of the torpedo boats and quite a number of the destroyers are still in existence and are giving good service, but the day of the fast running, lightly constructed, reciprocating engine as a main propelling unit for high-speed vessels has departed, never to return, either in the form of steam engines or internal-combustion engines. The steam turbine and the turbine with reduction gear have preempted this duty where power of any great magnitude is required.

THE MODEL TANK AND ITS EFFECT ON PROPULSIVE POWER.

At the period concerning which we are now dealing the Model Tank located at the Navy Yard, Washington, D. C., had been lately completed, and experiments on the models of hulls and the effect of varying lines of these hulls on their resistance to motion through the water were being carried out by Naval Constructor, now Chief Constructor, David W. Taylor, U. S. Navy. These experiments did not begin to bear fruit until the model of the *Connecticut* class of vessels was developed, when, with a hull of slightly 1,000 tons greater displacement than those of the vessels immediately preceding, the powers to drive at 18 knots were practically equal, while with the succeeding class of vessels of equal displacement with the *Connecticut* the power necessary for this same speed had been reduced by 2,000 I.H.P.

Investigations were also being carried on at the same time on the augment of resistance due to fitting the appendages, such as bilge keels, docking keels, propeller struts, etc., to the hull. It had been generally held that this augment was very

small, and in estimating power the usual rule, where Model Tank curves of resistance for the bare hull were given, was to multiply the bare hull tow-rope or effective horsepower by two to obtain the I.H.P. The result of this neglect of appendage resistance was to have many a good propeller condemned for inefficiency on account of its showing when compared with the bare-hull resistance, when in fact its efficiency has been very high as would be shown were the augment of resistance due to hull lines and location of propellers in regard to these lines and the augment of resistance due to the appendages taken into consideration. Another result was that the propulsive efficiency of the screw propeller was for years rated much lower than its actual value.

THE FIRST MODIFICATION OF THE BABCOCK AND WILCOX BOILER.

The Babcock and Wilcox Company after watching the performance of their new marine type of boiler in the *Chicago*, had devoted themselves to the task of correcting the existing faults by modifying the design, and in a short time developed a boiler which, as it was first used in the old sloop of war *Alert*, where it replaced the original Scotch boilers, was given the name "Alert type." This boiler, with only one minor modification, was the type with which we are the most familiar, as it won and held its place in our heavier vessels until the coming of oil fuel forced another change in the design which will be described later.

This new boiler had the steam drum located at the upper front angle with the front and back headers inclined down and back at an angle of 15 degrees to the vertical. The tubes between headers were inclined upward towards the back of the boilers at an angle of 15 degrees to the horizontal. The headers were sinuous, as before, but the front and back header faces were parallel to each other, the tubes entering the header plates normal to the surfaces and being expanded into them without belling of the ends. Belling was required in later specifications in order to prevent tubes withdrawing from the header plates

under stress due to possible deflection. The furnace sides were provided with steel boxes of square section providing water walls and reducing radiation losses, also making them much more durable under coal-firing conditions. The lower row of tubes directly over the fire were of 4-inch diameter and carried on them for about two-thirds of their length, from the front towards the back, a table baffle of fire tile making a gas-tight roof for this part of the furnace. The other baffling crossed the tubes and caused the gases to pass out of the furnace at the back and directed them across the tubes in three passes before passing out of the boiler into the uptake. The upper row of tubes carrying water and steam from the upper ends of the back headers to the drum, and thus completing the circulation chain, were also 4-inch diameter, and also carried a gas-tight baffle extending from the back to within about ¼ of their length from the front, there leaving an opening for the gas exit.

ADVANTAGES OF NEW TYPE OF BABCOCK AND WILCOX BOILER.

This boiler was eminently fitted for naval uses for it possesses the following advantages:

1. All tubes straight.
2. All parts accessible for cleaning and examination.
3. Easily repaired.
4. Ideal furnace conditions for efficient combustion.
5. Very efficient baffling.
6. High efficiency.
7. Comparatively slow falling off in efficiency at moderately high rates of forcing.
8. Stows well with minimum loss of space.
9. Any tube can be renewed without disturbance of others.

It has the following disadvantages:

1. Slightly unequal expansion between top and bottom of tube nest.
2. Possible tendency for circulation through lower tubes to be broken under high forced-draft conditions.

8. Large consumption of hand-hole gaskets.

The first and second of these disadvantages did not materialize to any noticeable extent under temperature conditions existing while coal was the fuel used by the Navy, and when oil became the fuel a still further change in the boiler was made in order to guard against the second. The third disadvantage is inherent with the boiler and cannot be entirely obviated but can be much reduced by use of graphite on the header bearing surfaces and care in removing the hand-hole plates.

DOUBLE-EFFECT EVAPORATORS.

During this same period the double-effect system of operating evaporators was installed on some of our vessels. In this system live steam is used as the heating medium in the coils of the evaporators of first effect and the steam generated by it in the shell of this evaporator is taken over into the coils of the evaporators of second effect, the drainage and uncondensed steam from these latter coils passing to the distilling condensers together with the steam generated in the shells of the evaporators of second effect.

Evaporators so connected operate with much greater economy than when operating in single effect, but this increase in economy is accompanied by a large decrease in capacity.

The evaporators of the period being all small for the demands made upon them, the reduction in capacity entailed by working them in double effect produced such shortage in the water supplies of the ships that the system was seldom used and soon lost favor, so that it disappeared from our vessels only to reappear several years later when the axiom that "contentment of crews breeds efficiency of ships" was more thoroughly appreciated.

THE APPEARANCE OF THE STEAM TURBINE IN THE FIELD OF MARINE PROPULSION.

The manufacture of the compound steam turbine, as it was first called, was begun in 1885, but its development was not

sufficiently great until 1894 to justify the expense of an experiment with it in the field of marine propulsion. In that year a syndicate was formed for this purpose and the experiment was inaugurated. After many delays and difficulties, not the least of which was the propeller phenomenon of "cavitation," a single-screw turbine engined vessel of small size, the turbine of the radial type of steam flow, was produced. The results obtained were unsatisfactory, so this installation of machinery was removed, and "replaced by three separate compound turbines, directly coupled to three screw shafts, working in series on the steam, the turbines being the high-pressure, intermediate and low-pressure, and designed for a complete expansion of the steam of 100-fold." . . . "The thrust of the propellers is balanced by steam pressure in the motors." This last statement indicates that in these new turbines the flow had become axial and that the turbines had assumed the form with which we are familiar today.

Both the original and the new installations were designed to produce 1,500 S.H.P. at 2,500 revolutions per minute.

Mr. Parsons, the inventor of the turbine, prepared an article on "The Application of the Compound Steam Turbine to the Purpose of Marine Propulsion," which was read before the British "Institution of Naval Architects" in 1897, and during the discussion of this paper the author stated: "Down to 15 knots screws should be driven direct; below that gearing would have to be used, and then the turbine would be applicable to all classes of ships. He looked with distrust on gearing; the small gearing might be made to work, but the application of large geared wheels was a questionable device."

The *Turbinia* made her initial bow to the engineering public in 1897, when she steamed down the line of British fighting vessels anchored at Spithead for the Queen's Diamond Jubilee, running at a speed stated to be 34½ knots.

The showing of the *Turbinia* was so impressive that the British Admiralty shortly contracted for the construction of

one destroyer, the *Viper*, fitted with turbine machinery, soon following it up with an order for the *Cobra*. The Parsons reaction turbine, as it became known, had come to stay as a marine propelling engine. In a short time the Admiralty had authorized its installation in a small cruiser, and in about 1905, laid down a battleship, the *Dreadnought*, with this same type of machinery. This latter vessel was far from being a success, as, in addition to having no cruising elements to give economy at low speeds, propeller troubles were experienced, due, probably, to too little clearance between the tips of her propeller blades and the hull.

In 1903, a possible rival of the Parsons turbine appeared in this country in the shape of an entirely new type of impulse turbine denominated as a "velocity compounded" turbine. This turbine, having four stages, was installed in a yacht named the *Revolution* and, in 1903, its operation was witnessed by a board of Naval Engineers who reported favorably upon its performance and recommended it for use in the Naval Service. This turbine today is known as the marine-type Curtis turbine.

Coincident with the arrival of the turbine in the marine field arose the demand for improved vacuum-producing apparatuses, on account of the high vacuums which were necessary in order to realize the full benefits of the new type of engines. The first arrivals in the field were the dry-air pump and the Parsons augmentor, both of these working in conjunction with the ordinary wet-air pump. Then appeared the Weir Dual pump, a combination of wet and dry pumps. The next were air pumps of the rotary type, water sealed, working in conjunction with a small rotary condensate pump. This apparatus as first installed was water sealed by the use of salt water taken from the sea and discharged overboard. The salt water carried away with it such a large amount of fresh water entrained as vapor in the air drawn from the condenser that a modification was necessary. This style of vacuum pump as installed today, uses fresh water as the sealing medium, this

fresh water operating in a closed circuit, passing through a tubular cooler after leaving the pump, the cooling being effected by means of salt water on the opposite side of the tubes. This type of pump is illustrated by the Leblanc pump. Later appeared Morison's Kinetic air pump, and, still later, the air ejector with rotary condensate pump. The latest comers in the field are the air ejectors using either high-pressure or auxiliary-exhaust steam, working in conjunction with the ordinary wet-air pump. The systems using the air ejectors in any of these combinations seems destined to drive the dry-air pumps and the augmentors completely from the field.

Variations in the forms of condensers also occurred, the prime object in these changes being to obtain the maximum condensing effect on the minimum of condensing surface and weight. Examples of these improved designs are the Morison "Contraflow," and Weir "Uniflow," and the Lovekin condenser which was designed by Mr. Luther D. Lovekin, late Chief Engineer of the New York Shipbuilding Company, and which is being installed in some of our new construction.

ADOPTION OF THE STEAM TURBINE IN OUR NAVAL SERVICE.

In 1905, reports received from abroad giving such glowing accounts of the performances of turbine-propelled vessels and of the advantages of this type of machinery over the old thoroughly tried out reciprocating type, the Navy Department determined to find out for itself, by experiment, the relative advantages and disadvantages of the two classes. In pursuance of this object three Scouts were authorized, one of them to have twin screws and reciprocating engines, one to have twin screws and Curtis turbines, and one to have four screws and Parsons turbines.

When completed, these vessels were put through a series of trials in competition with each other in order to distinctly mark out their points of relative superiority and inferiority.

The relative results of these trials were as follows:

1. Economy of propulsion: At speeds up to 20 knots the reciprocating-engine ship was superior to the vessel with Parsons turbines, and up to 21 knots was more economical than the vessel with Curtis turbines.

2. Overload capacity of engines: The Parsons turbines exceeded moderately the Curtis turbines in this respect and very much exceeded the reciprocating engines.

3. Reliability on trials: Both types of turbines took the lead over the light, high-speed reciprocating engine in this respect, the piston speed of the latter being 1,200 feet per minute.

The steam pressures, designed, were 250 pounds for the Parsons and 275 pounds for the Curtis turbines, with 275 pounds in the boilers of the reciprocating-engined vessel. No superheaters were fitted.

In actual service over long periods of time the answer to the question of reliability was reversed, the reciprocating engine, both for reliability and ease of repair, carrying off the prize, followed by the Parsons turbine and that, in turn, by the Curtis.

The Curtis turbine at this time was seriously handicapped by not having gone through the development and actual service experiences of its older rival, and it would have been miraculous should it so early in its career have beaten this thoroughly developed machine. For instance, one of the handicaps under which it labored was a mistaken idea as to the effect of steam in creating thrust in its passage through the turbine. The theory of the turbine being that all drops in steam pressure occurring in the nozzles, the pressure throughout the velocity stages remained constant and therefore no thrust due to differences in pressure could exist. Even if this were true there still remained the axial component due to changing the direction of flow of the steam as it passed through the turbine rotor blades. Whatever the cause, steam thrust did exist to an appreciable degree and its effects were serious in the port turbine where the direction of flow of the steam and the direction of thrust of the propeller coincided. In the starboard

turbine where steam flow and propeller thrust were opposed no trouble was experienced from this cause.

The Curtis turbines used in this case were all of the wheel-stage type with stages few in number. The next installation of this type of turbine occurred in a battleship, and with this second installation, the drum type of construction, with which the Parsons turbine had made us familiar, appeared for the first time. The number of stages was increased slightly over the number appearing in the Scout's turbines, but were not yet sufficiently numerous. This, together with the fact that the nozzles in service deteriorated so rapidly that the low economy was still further and seriously reduced, caused a reaction against the Curtis turbine for battleship use, and it did not again appear in this class of ship until several years later. When it did appear it was in a very much changed form from the earlier types.

Following this first turbine-propelled battleship, four others were built, all fitted with Parsons turbines with cruising turbines installed in order to obtain economy at low speeds ordinarily used in cruising and drill. These vessels were all very successful as measured by the standards set by turbine-propelled vessels. Luckily there were at hand data derived from a sister ship fitted with reciprocating engines which threw considerable light on the general subject of the propulsive efficiency of turbine-propelled vessels as obtained under the general practice of the day. These data demonstrated beyond question the superiority of the reciprocating engine with its relatively slow-turning propeller for battleship propulsion *in every particular.*

THE FINAL DESIGN OF HEAVY RECIPROCATING ENGINE.

In 1905 two battleships were authorized, and in the design of their engines three radical changes from existing practice were incorporated. In previous engines the distribution of power in the various cylinders had been extremely uneven, the I.P. cylinder generally developing a very heavy excess as com-

pared with the H.P. or the combined two L.Ps. In these new engines the ratio between the areas of the combined L.Ps. and the I.P. was much increased, the total ratio of expansion was greatly increased and, finally, the length of the main steam valves was made so great as to give short and straight steam passages from the valves to the cylinders. These engines were of 8,250 I.H.P., designed, at 1,000 feet piston speed, 265 pounds pressure in the H.P. valve chests, and about 50 degrees superheat, the piston speed, pressure and superheat being departures from previous vessels. This type of engine was repeated in a battleship of 1906, which engines were still further improved by fitting them with forced lubrication. This last item of change was of the utmost importance as in addition to conserving the lubricating-oil supply, it reduced shock on the engines, vibrations of the vessel, wear and tear on the journals and engines, initial friction and friction of the load, and increased the reliability. As expressed by one commanding officer in the Fleet, "the reciprocating engine without forced lubrication was one machine; with forced lubrication it was an entirely different one."

THE ADOPTION OF OIL FUEL FOR BATTLESHIPS.

Before this vessel was completed the Navy Department directed that she be equipped with oil-fuel apparatus and storage space be supplied for an auxiliary supply of oil fuel. This was done, the system of mechanical, that is pressure, atomization being supplied. This same system was also adopted for all classes of vessels, and as time has passed improvements have gradually crept in until, as oil-fuel burning stands today, no other system than this one of mechanical atomization would be considered.

TEMPORARY VICTORY OF RECIPROCATING ENGINES OVER TURBINES.

The reciprocating-engine battleship of 1906 having demonstrated its superiority very strikingly over the Curtis turbined

vessel of the same date and over the four Parsons battleships of the two succeeding years, it was decided that the two next vessels of this class should be engined with reciprocating engines. These vessels on their acceptance trials even bettered the performances of the preceding vessel having the same type of machinery, and the Bureau of Steam Engineering, in view of these performances, decided that no more turbine installations should be fitted in battleships until the turbine designers should so modify their practices as to aim at economy in propulsion, which would take into consideration the combined efficiency of turbines and propellers. According to their old practice they had aimed at economy of turbines only, and had based their promoting claims on these grounds.

THE DEVELOPMENT OF DESTROYER MACHINERY.

The building up of the destroyer flotilla, which had had no further increases since the first sixteen appropriated for, began in 1906. In deciding on the type of machinery to be adopted there was no hesitation in accepting turbines as the only desirable existing type, as experience with the earlier destroyers had fully demonstrated the unreliability of high-speed, light-weight, reciprocating engines for continuous high-power driving.

The first vessels built were all of three-shaft arrangement using Parsons turbines. Then some two-shaft arrangements using Curtis and Zoelly turbines were constructed. The two-shaft arrangement having demonstrated superior handling qualities in maneuvering the vessels, was finally accepted as the preferable arrangement, and in order to meet this condition, in addition to the Curtis and Zoelly turbines, a two-shaft Parsons arrangement was proposed by Mr. Charles P. Wetherbee, of Bath Iron Works, and was pronounced acceptable, although it had the serious drawback of only one main condenser.

DESTROYER CRUISING UNITS.

The conditions of service under which our fighting ships are required to operate makes it obligatory to conserve their fuel to the utmost extent, and as a long step towards such conservation all of the fighting ships of every class are fitted with such arrangements of machinery as will tend to reduce to a minimum the consumption of fuel at low speeds, and particularly at those speeds most frequently used in cruising and at drill.

As in the Parsons turbined battleships the three-shaft destroyers were fitted with a H.P. and an I.P. cruising turbine, one on each wing shaft. When the two-shaft destroyers first appeared, those having Curtis or Zoelly turbines, the economy at low speeds was obtained by using a decreased number of steam nozzles in the first stage. This method had also been used in the Curtis turbined scout and battleship. In later two-shaft arrangements where a certain fixed minimum cruising radius at a given speed was required, those vessels fitted with Curtis or Zoelly turbines were fitted with a little reciprocating engine on each shaft located forward of the turbines and exhausting into them, while in the two-shaft Parsons ships there was only one cruising engine located forward of the L.P. turbine and exhausting across the ship into the H.P. turbine on the opposite shaft. These engines were coupled to their respective shafts by couplings which could be quickly engaged and disengaged, some of them requiring the machinery to be stopped to do either, others requiring stopping only to engage, while still a third type was hydraulic and could engage and disengage with the machinery in operation.

FORCED-DRAFT BLOWERS DRIVEN BY ELECTRIC MOTORS OR BY STEAM TURBINES.

The reciprocating engines driving the forced-draft blowers on the earlier vessels both large and small, had given a great amount of trouble and were the frequent cause of unsuccessful

trials. In seeking for a more reliable power unit for these auxiliaries the electric motor had been called into service, and the first of such electric-driven blowers were installed in the 1905 battleships. These were so successful that they were fitted for some years in all subsequent battleships.

With the new destroyers, however, on account of the lack of electric power, some other form of power unit was necessary, and such a unit was found in the small high-speed steam turbine.

The first five new destroyers were coal burners, but all successive ones were oil burners, and it was with this fuel that the steam turbine established its undeniable superiority over the electric motor for forced-draft purposes. Having once thoroughly established this superiority it was not long before it had ousted the electric-driven blower from all service where it had before held undisputed sway.

REDUCTION GEARS, MECHANICAL AND ELECTRIC.

DIESEL ENGINES.

Mechanical Reduction Gears.—Your attention has already been called to the prophesy made by Sir Charles Parsons several years previous to this time, concerning the possible adoption of reduction gearing between the steam turbines and the propeller shafting in order to make the turbine suitable for the propulsion of slow-speed vessels. Both Sir Charles, in England, and the late Rear Admiral Melville, with an associate, Mr. Macalpine, had been working for several years on the problem, and each reached his own solution at about the same time. Sir Charles exposed his invention on board the old merchant steamer *Vespasian*, while the Melville-Macalpine invention, which had been taken up by the late Mr. George Westinghouse, was disclosed through the medium of a trial unit built at the works of the Westinghouse Machine Co. The main difference between the two types of gear was in the features of preserving alignment between the pinions and gear wheel, which in the Parsons gear was established by rigidity

of foundation, while in the Melville-Macalpine gear the pinion line was given a certain amount of play by means of a flexible I-beam support, afterwards changed to an hydraulic cylinder support, and later, reverting again to the I-beam, with a flexible shaft connection to the turbine shaft.

Electric Reduction Gear.—About 1908, while the machinery plans for the battleships of that year were being developed, a Mr. Fessenden presented through the President of the United States, a proposition to install electric propelling machinery for those vessels. The plan submitted was very sketchy and contained several very objectionable features which, combined with the fact that such radical departures as the adoption of an entirely untried power unit for the propulsion of capital ships is never done by the Navy Department, caused the scheme to be rejected. The following year, Mr. W. L. R. Emmett submitted a proposition for electric propulsion for the battleships of that year. Mr. Emmett's scheme was much better developed than that submitted by Mr. Fessenden but was not adopted, as it was considered that the experiment, if carried out, should be carried out on a less important unit of the Fleet than a battleship.

Diesel Engine.—For several years previous to the date of which I am speaking Dr. Diesel, a German citizen, had been experimenting with an internal-combustion engine to use heavy oil as a fuel in place of the lighter oils which were then generally used. It is said that this engine was first designed with the idea of using powdered coal as fuel. In 1897, Colonel Meier, representing Mr. Adolphus Busch, of St. Louis, read a paper before the annual meeting of the American Society of Naval Engineers, describing this engine in very glowing terms and advocating its more general use.

By 1911 this engine had come into general use as the main propelling units for surface propulsion of submarines and was rapidly gaining ground in the field of low-powered installations for slow-speed merchant craft. Many men in the marine engi-

neering world were influenced by the attractiveness of the proposition and by the untruthful statements of unprincipled promotors and were dreaming of the disappearance of steam engines of all types with their necessary boilers, and of their replacement by units of the new type of engine for all powers. Our Service adopted the Diesel engine for submarines in place of the gasoline engine which we had been using since the revival of the submarine in 1900.

TRIALS OF MECHANICAL AND ELECTRIC REDUCTION, AND OF THE DIESEL ENGINE, FOR LARGE VESSELS.

In 1910 opportunity offered for testing out mechanical gears; in 1911, electric gears. In 1912, some of our naval engineers being severely bitten by the Diesel bug and the pressure being exerted by the Diesel engine interests having become severe, it was determined to give this form of engine its opportunity also.

The collier *Neptune* was selected as the vessel to be fitted with mechanical gears, the machinery being manufactured by the Westinghouse Machine Company.

The collier *Jupiter* was chosen as the field in which the ideas of Emmett and the General Electric Company were to be developed and demonstrated.

The fuel ship *Maumee* was set aside for the Diesel engine experiment, plans for two-stroke cycle engines of Nurnberg design being obtained from the Electric Boat Company while the actual work of construction was performed at the New York Navy Yard.

The trials of the *Neptune* and *Jupiter* demonstrated fully the suitability of both types of reduction for naval purposes. Of the *Maumee*, the best that we can boast is that we possess the highest powered Diesel engine installation afloat up to the present time.

The three vessels were all twin screw, while the total S.H.P. of the *Neptune* and of the *Jupiter* were about 7,000 and that of the *Maumee* 5,000.

OIL FUEL FOR LARGE VESSELS. THE TURBINE DEFEATS THE RECIPROCATING ENGINE. REDUCTION GEARED CRUISING ELEMENTS.

In 1911 were developed plans for battleships which were epoch making, as these vessels were designed to use oil as the only fuel while one of them demonstrated that the Curtis turbine had passed from the development stage with all its troubles and disheartenings, and had reached the state of a completely successful marine turbine.

The other vessel was fitted with reciprocating engines of the now thoroughly standardized type, and as the two vessels offered again an opportunity for a thorough test between the two types of machinery, that is, reciprocating engines and turbines, every means were taken to give each its maximum efficiency excepting superheat which, for a good reason, was omitted in the designs.

The clearances in the cylinders of the reciprocating engines were reduced to a minimum, steam and exhaust velocities were reduced, forced lubrication was fitted, and economy of auxiliaries at low powers was provided for as much as possible.

In the design of the turbines for the other vessel, the revolutions at full power had been reduced from 280 to 220, the diaphragms between stages were split instead of being solid discs, the blading was made heavier and the design of the root was changed to a stronger and more reliable form. The compound Curtis turbine made its first appearance in our Service, each shaft, of which there were two, being fitted with two turbines, an H.P. and a L.P., the H.P. being of the multiple-wheel type while the L.P. was of the drum type with impulse blading.

ADOPTION OF GEARED CRUISING TURBINES.

With this turbine ship also appeared the geared cruising element on large ships, it having been installed on this vessel after being adopted for the succeeding vessel of the battleship class. This type of cruising element had been already adopted for destroyers where its use had become imperative on account of

the demand for an increased cruising radius at an increased speed over that of the earlier boats.

The trials of these two vessels and of the succeeding one, which was also fitted with compound turbines and reduction-gear cruising element, but which had its power distributed on four shafts, resulted in the complete defeat of the reciprocating engined ship at all speeds except the maximum, and this defeat, in combination with the abnormal increase in powers which occurred shortly after, resulted in the complete elimination of the reciprocating engine from consideration for propelling purposes in vessels of high speed and power.

FINAL MODIFICATION OF BABCOCK AND WILCOX BOILER. MODIFICATION OF BOILERS OF THE EXPRESS TYPE.

With the adoption of oil as the only fuel for large vessels, the Navy Department considered that some modifications should be made in the form of Babcock and Wilcox boilers if they were to be used under these fuel conditions. These wishes of the Department were at once met by the Babcock and Wilcox Company, which developed at once a new boiler conforming to them.

These changes were as follows: (1) As a matter of course, all grates disappeared; (2) side water walls were eliminated, heavy fire brick walls taking their place; (3) angles of tubes and headers were changed, tubes being inclined 18 and 20 degrees to the horizontal and inclination of headers being changed accordingly; (4) down-comers between steam drum and lower front cross-box were installed; and (5) lower row of 4-inch tubes were done away with and two rows of 2-inch tubes were installed in place of them. The first of these boilers was installed and thoroughly tested out at the Oil Fuel Plant and met every expectation.

Several years later boilers of the express type having water drums of an oval form were modified by changing from the oval form to the cylindrical form of drum. This oval form of drum had given trouble from time to time in express boilers

having curved tubes, and in the Yarrow type with straight tubes under the arduous conditions of war service.

ADOPTION OF REDUCTION GEARS, MECHANICAL AND ELECTRICAL, FOR HIGH POWERS.

The trials of mechanical reduction gears having been satisfactory, an installation as the main power units in a destroyer was authorized in 1913; the results of this installation were so gratifying that the direct-connected turbine is no longer considered, mechanical gearing and small diameter, high-speed turbines having driven them completely from the field in such vessels as destroyers and scouts where high speed on minimum weight are required and where protection for such machinery is impossible.

Our electric reduction experiment having resulted so successfully, it was decided, in 1914, to apply it to a battleship. In such vessels and vessels of the battle-cruiser type, in addition to speeds varying from moderately to very high and with high to very high powers, the prime requisites are as low a weight as possible combined with reliability, economy at low speeds, simplified and reduced piping arrangements, ease in overhaul and supervision and maximum possible protection of ship and machinery against serious crippling from gun-fire and torpedo attack. These being obtained in a much greater degree with electric than with mechanical reduction gear, the Navy Department, against much opposition from outside interested parties, adopted the electric reduction for all heavy fighting ships, and stands on that decision today.

In the fifty-year period covered by this review steam pressures have risen from 30 pounds to 250 and 300 pounds per square inch, piston speeds of reciprocating engines from 240 and 250 feet to 1,000 for heavy and 1,200 for light engines, weights have fallen from a very few horsepower per ton to 30 per ton for heavy ships and 50 per ton for light, very fast ones. The true types of water-tube boilers have driven the box type, the Scotch type, and the semi-water tube, such as the old

Martin boiler, from the field. Sail power has disappeared and steam power is king. Powers have risen from a few hundred horsepower to 28,000 for destroyers, 60,000 for battleships, 90,000 for scouts and 180,000 for battle cruisers. The reciprocating engine has practically disappeared, its place being taken by the direct-connected steam turbine, and this latter, in turn, being forced to give way before the superior qualifications of mechanical and electric reductions. Notwithstanding all of these changes we find ourselves today using gearing between the main engines and the propeller shafts, water-tube boilers and superheaters as did Commodore Isherwood in the famous *Wampanoag* of fifty years ago. Machinery and materials have improved, but otherwise the ideas of this famous engineer, ideas for which he was criticized and condemned, still prevail and he receives the honor at this late date which he should have received in the days of his accomplishment.

In the years of endeavor that have passed since the rebuilding of the Fleet was begun the Naval Engineers have been most ably supported by a body of men who, while not in the Service, are decidedly of it, as their technical life for years has been so closely identified with naval work. On these men devolved the task of developing the details of the machinery installations and the production of actual in place of paper machinery. These men, whose talents have been devoted to the interests of the Service and whose names should be found in any history of the development of our Navy which may be written in the future are George W. Dickie, late Chief Engineer of the Union Iron Works; Horace See, N. P. Towne and John F. Metten, successively Chief Engineers of Cramps Ship and Engine Building Company; Luther D. Lovekin, late Chief Engineer of the New York Shipbuilding Corporation; Charles F. Bailey, Chief Engineer of the Newport News Shipbuilding and Dry Dock Company; R. Warriner, Chief Engineer of the Fore River Shipbuilding Company; and Charles P. Wetherbee, of the Bath Iron Works. These men should all have their names appear on the roll of honor of the period.

The Nation is now at war. The Navy is undergoing the acid test. We may all rest fully assured that when the crucial hour of supreme trial arrives the engineers and engineering material of the Fleet will not be found derelict in their duty of upholding the best traditions of the Navy and the honor of the Flag.

NOTES ON THE SUBJECT OF BALANCING.

By N. W. Akimoff.*

INTRODUCTION.

Commander Cleary's interesting communication on *Dynamic Balancing* in the February issue of this Journal is no doubt a good sign: at last the subject of balancing seems to be coming into its own, being, as I always maintained, a separate branch of mechanical engineering, and not a shop method at all; and it is not a very difficult subject, provided that it is handled in a correct manner, and with the full conviction that there is not another branch of engineering in which theory and practice are in closer agreement than here.

Commander Cleary most courteously mentions my name in connection with the matter, but I fear that I must have been somewhat unsuccessful in driving home the main points of the subject, because I evidently was misunderstood. A machine, such as is described in Commander Cleary's article, could not possibly have been built by anyone that really took the trouble to absorb the meaning of my short communications to the A. S. M. E. It is absolutely necessary to put things straight for fear that someone else might use the machine and the method described by Commander Cleary, and great may be his disappointment at the result! The danger of Commander Cleary's paper lies in the unquestionable sincerity and ability of its author; it is a step in the right direction, and made with the best intentions, but from entirely wrong premises, as I hope will be presently made clear. In what follows reference to the paper in question will be made by pages.

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DEFINITIONS.

In establishing the correct premises it is absolutely necessary to review and straighten out the usual definitions. In writing this article for the naval engineers, who as a class are of a rather high standard, I will use a little different language in defining things than in addressing a body of "practical men" (so called).

By *static balance* is meant the condition when the center of gravity of the body lies exactly on the axis of rotation. Analytically,

$$\bar{y} = \frac{\Sigma my}{M} = 0; \quad \bar{z} = \frac{\Sigma mz}{M} = 0;$$

where $\bar{y}$ and $\bar{z}$ are the coördinates of the center of gravity.

Static unbalance is measured in two manners: either by stating that the center of gravity is *so far* out, in which case it is necessary to specify, in each case, the mass of the body; or, by specifying simply the product of the mass (usually weight) by the distance of the center of gravity from the axis; this product is usually given in *inch-ounces*, although it could also be expressed in foot-pounds, or any other units of this kind. If this value, say $\triangle$ M is multiplied by ω^2, the result is the so-called centrifugal force, due to unbalance; but it should be remembered that a body whose center of gravity is *off the axis* is unbalanced anyhow, whether it is run or not; the running only emphasizes the effect of unbalance. Static balance *positively can not be obtained on the ways*, except if the body is light or the journals very small. In my laboratory accurate records are kept as to the discrepancy of the results, in inch-ounces, as obtained on the parallels from those obtained on an accurate machine. In my last communication to the A. S. M. E. I proposed tentative formulae and charts by which this *sluggishness* can be predicted, more or less.

By *dynamic balance* is meant the condition $\Sigma myx = o$, and, at the same time, $\Sigma mzx = o$; in other words, that the products of inertia vanish; geometrically, this means that one of

the principal axes of the corresponding momental ellipsoid is exactly coincident with the axis of rotation, x; otherwise it will be somewhat inclined to the latter. The dynamic unbalance is being measured, as proposed by the present writer, in inch-ounce-inches; multiplying this by ω^2 we have the value of the corresponding centrifugal couple. It makes absolutely no difference whether the body is running or not; if the momental ellipsoid is not straight the body is unbalanced dynamically. Take, for instance, a three-throw crankshaft, not provided with counterweights; it is an easy matter to draw a momental ellipsoid and to ascertain the fact that it will not lie symmetrical with the axis of rotation. However, the running *may* manifest the fact that the body is out of balance, if the speed is high, or the supports are of the nature especially well adapted for visibly responding to the effect.

From this standpoint the extract from the general specifications of the U. S. Navy can only lead to confusion (p. 3, JOURNAL A. S. N. E., Feb., 1918), because, in the first place, *no perfect static balance can be obtained on knife edges;* and, second, because the operation under steam in its own casing may exhibit vibrations due to either static or to purely dynamic unbalance, or to the mixture of the two; so that, apparently suggesting a *method* of testing, this paragraph is absolutely misleading; the stipulation as to the excess speed of 25 per cent of the normal is even worse, and will be referred to later. The correct way would be to say: "Great care will be taken to insure the rotor being balanced statically, the tolerance of the static unbalance being not over * * * inch-ounce; then to balance the rotor dynamically, the tolerance of dynamic unbalance not exceeding * * * inch-ounce-inches." (Such tolerances to be standardized.)

ERRONEOUS DEFINITIONS.

On page 5 of the paper referred to Commander Cleary makes certain assumptions which are not quite correct:

(*a*) In the case of a long rotor, static balancing does not

always increase the dynamic unbalance (think always of the momental ellipsoid); in my laboratory it is a fixed rule, from which no departure is allowed, that the correction for static balance must not in any way alter the longitudinal position of the transverse plane through the center of gravity. In this manner the dynamic unbalance is either decreased by static balancing or, at least, is left exactly the same as before; of course introducing the correction *anywhere at all* would most likely increase the value of dynamic unbalance. What Commander Cleary meant is that by proper location of the static correction it would be possible to *decrease* dynamic unbalance, if only the offending point were *one*, located not necessarily in the transverse plane through the center of gravity. With this I agree absolutely, but there is no such *one* heavy point, as a rule, but static unbalance is due to the fact that the center of gravity of the whole body is a little off the axis.

(*b*) It is not the fact that the *rotor* has any critical speed that counts, but the fact that the *supports* happen to have a free period nearly coincident with that of the actual rotation; remove the disturbing cause and the forced oscillation cannot take place. This is what balancing means. Of course, the rotor itself may have a "critical" speed of its own, but this is a matter so entirely foreign to the subject of balancing that it is best not to bring it here at all.

(*d*) What is meant, probably, is the fact that all *mechanical* energy for rotating the body must come from *within* the oscillating system and not from outside. The idea is most important, of course, but it most certainly was not carried out successfully in the flexible coupling, Plate I, which is about the most unfortunate arrangement one could think of; if the coupling were on the other end this would improve things a bit, although a flexible coupling should never be used in matters of this sort.

(*e*) It is not the existing vibration that should be magnified—there is no information that can be gained thereby;

but its actual amplitude must be *increased* so that it will become visible even without any elaborate magnification; in other words, it is absolutely imperative to make the free vibration of the supports synchronous with the rotation, or vice versa. On this subject a few words will be said a little later.

(*f*) This only applies to a rational apparatus: in a home-made machine this may or may not lead to results.

On page 7 dynamic balance is defined as a condition in which the centers of gravity of all elements lie on the axis of rotation. That this definition is wrong can instantly be seen by reference to a body such as a six-cylinder crankshaft.

WHY THE SUBJECT DOES NOT TAKE ROOT.

The subject of balance would be better understood and taken up by a greater variety of engineers if it were not for the main obstacles: one lies in the fact that anyone who made up his mind to build a balancing machine actually thinks he can do it; he can, but only if he has mastered the elements of dynamics to start with. *Horse-sense* alone is here absolutely helpless; the rational premises from which to start, and from which a very great variety of new types of balancing machines can be derived, will be established presently. The second obstacle centers in the fact that anyone seems to be easily convinced by the manufacturer's assurance that the article has been balanced on a special machine, by a special man, in a special department, etc. Call on any chief engineer of any shipyard or large shop and he will tell you the same story: "Never have had any trouble like so many other concerns seem to. We have a special expert who attends to it. We don't know how he does it, but *he just does*. The public buys our goods, the Government passes on them, so what else do we want?"

If it were only known how helpless are the big corporations in matters of balance, and how unwilling to confess their ignorance unless absolutely forced to, the public and the Gov-

ernment would change their attitude and demand *standardized balance,* with definitely established tolerances, secured on a *rational* apparatus, the action of which is clear and positive and requires no "assumptions" or special "theories."

ELEMENTS OF THEORY OF BALANCING MACHINES.

We all know the equation of damped oscillations:

$$M\frac{d^2a}{dt^2} + D\frac{da}{dt} + F\,a = 0,$$

where M is the generalized mass and a the generalized coördinate. Its integral is $a = a_0\, e^{\lambda t}\, sin\omega t$, where $\lambda = D/2M$ and $\omega = 2\pi/T$ the period being $T = \frac{4\pi M}{\sqrt{4MF - D^2}}$.

We also know that the ratio of these consecutive deflections, on the same side, is $a_1/a_2 = e^{\lambda T}$.

Developing the exponential and dropping higher terms we have $a_1/a_2 = 1 + \lambda T$; so that

$$(a_1 - a_2)/a_2 = \lambda T = DT/2M = \beta, \textit{ say};$$

in other words, the quantity β measures the relative decrease of the amplitude. Now, then, if the oscillation were an undamped one the period would naturally be $T_0 = 2\pi\sqrt{\frac{M}{F}}$.

The ratio of the two periods will be

$$T/T_0 = \frac{1}{\sqrt{1 - D^2/4MF}} = 1 + D^2/8MF = 1 + \frac{D^2T_0^2}{32\pi^2M^2};$$

in other words,

$$T/T_0 = 1 + \frac{\beta^2}{8\pi^2}.$$

It will thus be seen that even if β is very considerable, such as 20 per cent. still, the difference between the two periods will be practically nothing. Hence the important result, that whenever the damping is proportional to the first power of velocity, the period is not greatly different from that of undamped motion, unless damping is excessive.

The next step is to consider the action of some harmonic force (generalized) on such a system. The equation of such a *forced* oscillation will then be

$$M\frac{d^2a}{dt^2} + D\frac{da}{dt} + Fa = E\,sin\,\omega t,$$

the period of the imposed force being, of course,

$T_t = 2\pi/\omega$ seconds.

The solution consists of two parts: the *complementary function*, in which we are not in the least interested, because it represents that part of the motion which soon dies out (on account of the factor $e^{-\lambda t}$, as has just been shown); and the *particular integral*, which represents the forced oscillation proper

$$a = a_0\,sin\,(\omega t - \epsilon),$$

where $a_0 = \frac{E\,sin\,\epsilon}{D\omega}$; and $tan\,\epsilon = \frac{D\omega}{F - M\omega^2}$.

Here the appearance of the constant ϵ suggests a certain *lag* of the effect behind the cause.

What interests us most is the question of energy loss due to frictional work of the damping. The elementary work of the damping force (generalized) is evidently $= D\frac{da}{dt}da$.

Integrating this over a complete period of oscillation we have (since the constance of the amplitude means that *all* energy goes directly into frictional work) the time integral of energy for period

$$= \omega^2 a_0^2 D \int_0^T cos^2\,(\omega t - \epsilon)dt;$$

which easily integrates into $\frac{\pi E^2 sin^2\epsilon}{D\omega}$.

The period being $= 2\pi/\omega$, we get the mean energy,

$$W = \frac{E^2 sin^2\epsilon}{2D}.$$

This expression is maximum when $\epsilon = 90°$ (lag of effect behind the cause); which (from the above expression of $tan\,\epsilon$) also means that

$$F = M\omega^2 \text{ or } \omega = \sqrt{F/M},$$

that is, the period of the imposed force is equal to the free period of the same system *without damping* ($D = 0$); we

have just seen, however, that this period does not materially differ from the actual free period, if only damping is not excessive.

Now, since the condition of maximum energy is $\varepsilon = 90°$, this means that the equation of the oscillation $a = a_0 \sin(\omega t - \varepsilon)$ will now be

$$a = a_0 \cos \omega t = \frac{E}{D\omega} \cos \omega t,$$

which means pendular motion with an amplitude $E/D\omega$, which can become quite considerable if the damping is small. However, this amplitude can never reach infinity, no matter how small the value of D, because with the increase of amplitude the restoring forces will be no longer proportional to deflections, a, and the fundamental equation will not hold true. Hence the absurdity of the supposition that a fiddler can fiddle down a suspension bridge, etc.

SYNCHRONISM.

Still, if the period of the imposed force is near or equal to the free period of the system, both the energy and the amplitude are considerable, and such a condition, called *synchronism,* will necessitate a few remarks.

To begin with, synchronism and resonance mean two things entirely different: synchronism means that a system is acted on by an external agency whose period happens to coincide with the free period of the system, but, in itself, is unchangeable. Take, for instance, a rolling ship; its free period may be exactly the same as that of the waves, but the latter is not subject to change on that account. Now by resonance is meant a condition by which both periods, that of the free oscillation and that of the external agency, *interact,* each being influenced by the other. The wobbling table of Dr. Sommerfeld is precisely in that class; this is really a problem of two degrees of freedom; the motor acts on the table and the table acts on the motor. The loss of efficiency in the destroyers with unbalanced turbines also belongs to the same

order of things. It is also of interest to note that the curve of the amplitude increase in the vicinity of synchronism is a bell-shaped affair, as roughly shown in Fig. 1. If, for instance, the synchronous speed is 400 r.p.m., the amplitude decreases very quickly on either side of it, and if the speed is 25 per cent higher or lower the oscillations may vanish, apparently, although the strain on the bearings and the loss of power may be considerable.

Now, having in mind the fundamental equation given above and the principle of synchronism, one can develop any number of designs of rational balancing machines. Starting without this background one is quite sure to fail. Let us analyze the requirements of a balancing machine from the standpoint of the theory briefly outlined above.

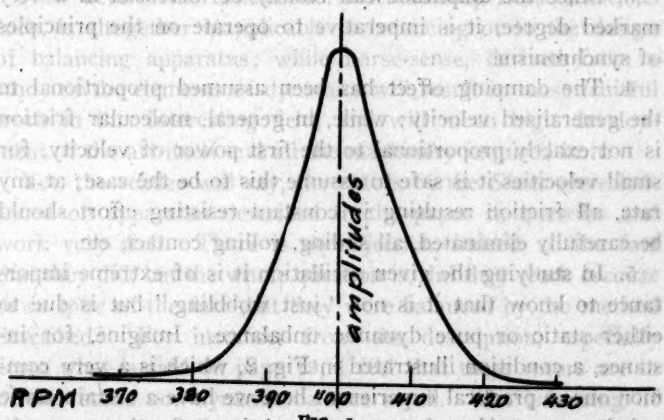

FIG. 1.

DEDUCTIONS FROM FUNDAMENTAL EQUATION.

1. All methods based upon the marking of "high spots" are absolutely worthless; they are all based upon the value of the angle Σ of phase-difference; now this angle depends upon the speed, ω, and therefore does not present anything tangible that can definitely be located. Reversing the rotation

is likewise a questionable method, since it is not at all an easy matter to secure exactly the same speed as before. Furthermore, such a high spot, even if the speed could be accurately determined and maintained, would not be of any positive value that can be interpreted numerically.

2. What we called E, that is the generalized acting force, must necessarily be due to the unbalance itself; all outside influences should be paralyzed, lest they, too, should create oscillatory motion; flexible couplings should be banished from serious tests.

3. Since the amplitude can readily be increased in a very marked degree, it is imperative to operate on the principles of synchronism.

4. The damping effect has been assumed proportional to the generalized velocity; while, in general, molecular friction is not exactly proportional to the first power of velocity, for small velocities it is safe to assume this to be the case; at any rate, all friction resulting in constant resisting effort should be carefully eliminated, all sliding, rolling contact, etc.

5. In studying the given oscillation it is of extreme importance to know that it is not "just wobbling," but is due to either static or pure dynamic unbalance. Imagine, for instance, a condition illustrated in Fig. 2, which is a very common one in practical experience: here we have a certain static unbalance, manifested as centrifugal force S; also a certain

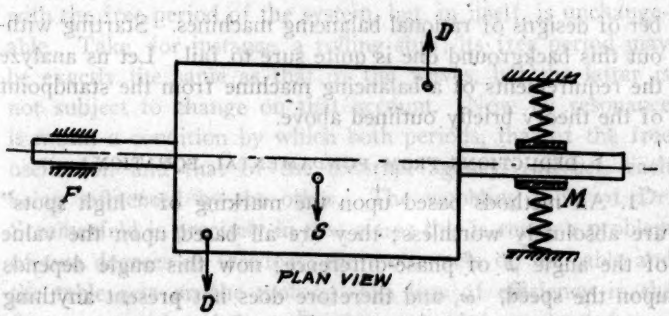

FIG. 2.

amount of dynamic unbalance, evidenced by the centrifugal couple D–D; F is the fixed bearing and M the moving bearing with spring adjustments. Now if the couple due to the centrifugal force S, taken with reference to the fixed bearing, F, is exactly balanced by the centrifugal couple D–D, the bearing M will exhibit absolutely no effect, and we might naturally suppose that the body is in most perfect balance. An ignorant engineer will often feel quite happy over securing just such a result, feeling that *one end is now all right*, not realizing that his whole foundation is radically wrong.

Now here is the conclusion: machine-shop horse-sense, as such, is altogether inapplicable to balancing or to the design of balancing apparatus; while horse-sense, derived from a study of the elements of dynamics will result in most fruitful work in this interesting field, and will mean high development of the art itself and all the machinery required in it.

Yet, the reader will say, the rotors of the *Shaw* were balanced on apparatus described in the paper in question and work very well. There is no objection, indeed, to the fact that they did, but the whole question is, did they work *because* or *in spite* of the fact that they were balanced by the method described? For instance, from the description, especially from what is said on page 19, I feel that these rotors are in neither static nor dynamic balance at all. The balancing of the propellers, outlined on page 17, is especially significant: anyone familiar with the first principles of balancing will at once see why it was necessary to take off 30 pounds and then immediately to put it back again; also why the second propeller, not balanced statically, ran just as well as the other. Just think of a body, balanced dynamically, but out of balance statically!

When in a matter of this sort it is possible for two engineers to disagree quite as completely as here (I agree with Commander Cleary only in two points: the *importance* and the *object* of balancing; but I, most respectfully, disagree with him as regards the method, the apparatus used by him, his defini-

tions, conclusions and his results), then it is clear that something is radically wrong; it becomes, therefore, necessary to have the subject looked into more thoroughly with the immediate purpose of (*a*) establishing the correct basis on which to work, and (*b*) adopting certain limits of accuracy (tolerances), thus establishing the very desirable uniformity of the product.

It appears that I am to be invited to present a paper on balancing apparatus before one of the best known scientific organizations of our land some time in December. Let us hope that in discussion many ingenious balancing machines may be proposed by the readers of this short article, and that all of these will be of the rational kind, based on the simple differential equations given above.

AIR PROPELLER PERFORMANCE AND DESIGN BY THE SPECIFIC-SPEED METHOD.

By M. C. Stuart,* Associate.

The purpose of this paper is to develop and illustrate a new method of treatment of the performance and design of geometrically similar air propellers. In the methods used at present expressions for torque and thrust coefficients are derived from the known laws of propeller performance based on the theory of similitude. The values of these coefficients are determined for various values of the slip function and are used to determine the characteristics of geometrically similar propellers for the same values of slip function (see references *a* and *g*).

The Specific-Speed method of treatment of the subject is an adaption and extension of methods which have been successfully applied to hydraulic turbines, centrifugal fans and centrifugal pumps (see Appendix III). In the more usual type of problem arising in the application of a propeller to a machine the requirements of power and forward velocity of the machine are known, and the diameter or rotational speed of the propeller, or both, must be determined to meet these requirements. In the Specific-Speed method the rotational speed and diameter of the propeller are expressed in terms of power and forward velocity, and a direct method of treatment is thus obtained. The formulae required will first be developed from the laws of similitude for the performance of geometrically similar propellers; the relation between the various units will be shown in the form of curves; and, finally, examples will be given of the use of the method in the solution of problems.

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DERIVATION OF FORMULAE.

The laws for the performance of geometrically similar propellers, as affected by changes in diameter and speed, have been developed theoretically, and have been proven experimentally to be as follows: For a constant diameter of propeller, and at constant efficiency operation, the forward velocity is proportional to the rotational speed, and the power developed is proportional to the cube of the speed. Also, at constant rotational speed, and for constant efficiency operation, the forward velocity is proportional to the diameter, and the power developed is proportional to the fifth power of the diameter (see references *a* and *b*). These laws may be expressed by the following two equations:

$$V = C_1 N D, \quad . \quad . \quad . \quad (a)$$

$$P = C_2 N^3 D^5, \quad . \quad . \quad . \quad (b)$$

in which

V = Velocity of forward movement of propeller relative to still air.

P = Power reduced to standard air density.

N = Rotational speed.

D = Diameter.

Eliminating D and solving for N in these equations we have:

$$N = C_3 \frac{V^4}{P^1}, \quad . \quad . \quad . \quad . \quad (c)$$

If, in equation (*c*), V and P be taken as unity, C_3 will be the speed required to produce unit velocity with unit power, and this speed is termed the specific speed and designated by the symbol N_s, then

$$N = N_s \frac{V^4}{P^1}, \quad . \quad . \quad . \quad . \quad (1)$$

or

$$N_s = N \frac{P^1}{V^4}, \quad . \quad . \quad . \quad (1a)$$

Eliminating N from equations (*a*) and (*b*) and solving for D, we have:

$$D = C_4 \frac{P^{\frac{1}{5}}}{V^{\frac{1}{2}}}, \quad . \quad . \quad . \quad . \quad (d)$$

If, in equation (d), V and P be taken as unity, C_4 will be the diameter of propeller required to produce unity velocity with unit power, and this diameter is termed the specific diameter and is designated by the symbol D_s; then

$$D = D_s \frac{P^{\frac{1}{5}}}{V^{\frac{1}{2}}}, \quad . \quad . \quad . \quad . \quad (2)$$

and

$$D_s = D \frac{V^{\frac{1}{2}}}{P^{\frac{1}{5}}}, \quad . \quad . \quad . \quad . \quad (2a)$$

We may now define the "Specific Speed," N_s, of a propeller, operating at any given efficiency, as the speed of a geometrically similar propeller which is of such a diameter D_s as to produce unit power at a unit velocity and with the same efficiency. Also, the "Specific Diameter," D_s, of any propeller, operating at a given efficiency, is the diameter of a geometrically similar propeller, which, when running at a speed N_s, will produce unit power at unit velocity and with the same efficiency.

Designating the tip speed of the propeller by v, and the specific tip speed by v_s, we have:

$$v = \pi DN, \quad . \quad . \quad . \quad (e)$$

and

$$v_s = \pi D_s N_s = \pi D \frac{V^{\frac{1}{2}}}{P^{\frac{1}{5}}} N \frac{P^{\frac{1}{5}}}{V^{\frac{1}{2}}},$$

$$v_s = \frac{\pi DN}{V}, \quad . \quad . \quad . \quad (3)$$

or from (e) and (3) we have:

$$v_s = \frac{v}{V}, \quad . \quad . \quad . \quad . \quad (3a)$$

and

$$v = v_s V, \quad . \quad . \quad . \quad . \quad (3b)$$

The specific tip speed, v_s, of a propeller, operating at any given efficiency, may be defined as the tip speed of a geo-

metrically similar propeller required to produce unit forward velocity at the same efficiency.

RELATION TO SLIP AND SLIP FUNCTION.

Inasmuch as the methods in use at present involve the use of slip and slip function, the relation between these and the specific units may be noted.

From the definition of slip,

$$V = pN(1 - s) = mDN(1 - s), \quad . \quad (f)$$

in which

p = pitch,
s = slip,
m = ratio of pitch to diameter.

From (3), above, $v_s = \frac{\pi DN}{V}$,

substituting in (f)

$$v_s = \frac{\pi}{m(1 - s)}, \quad . \quad . \quad . \quad . \quad . \quad (g)$$

and

$$s = 1 - \frac{\pi}{mv_s} \quad . \quad . \quad . \quad . \quad . \quad (h)$$

also, the relation between the slip function, V/ND and v_s is given directly by the expression,

$$\frac{V}{ND} = \frac{\pi}{v_s} \quad . \quad . \quad . \quad . \quad . \quad . \quad (i)$$

The use of slip and slip function is not necessary in the Specific-Speed method, but the relation between slip, slip function and the specific units may be determined at any time, if desired, by the use of equations (g), (h) and (i), provided suitable units are used.

UNITS.

In the development of the formulae to this point no units have been mentioned, and a homogeneous system may be considered to be used. However, in the application of the formulae to problems, it is probably desirable to select commercial units which lead directly to desired answers. Inas-

much as the values of specific speed and specific diameter are defined in terms of unit velocity and unit power it is necessary to first define these units. It is proposed to select for this purpose a unit velocity of 100 miles per hour and a unit power of 100 horsepower. These units of velocity and power lead to values of N_s and D_s, which are of the same degree of magnitude as the speed and diameter of actual propellers. In fact, when using these units, the "specific propeller" may be considered as one having a speed and diameter which will produce a velocity of 100 miles per hour with a power of 100 horsepower.

As regards horsepower, either the useful (thrust) horsepower developed by the propeller or the effective (torque) horsepower produced by the engine may be used. It is considered to be more convenient to use the thrust horsepower developed by the propeller in the specific speed formulae, and this power is therefore used in the present work. The effective horsepower may be determined at any time from the knowledge of the values of efficiency and useful horsepower. As will be shown later, the Specific-Speed method may be used to solve any type of problem involving power, velocity, speed and diameter. With these known, torque and thrust may also be determined by the usual formulae which express the relation between power, torque and thrust.

DENSITY OF AIR.

The effect of air density need not be considered in any of the exponential formulae developed in the Specific-Speed method. Since at any diameter and speed the power is proportional to the air density, the actual power may be reduced to a power at any standard air density by simple proportion. This is essentially the treatment for density used by the present methods. In this work a standard density of 0.0750 pound per cubic foot is used in developing values of N_s and D_s, and the values of N_s and D_s apply for that standard density.

APPLICATION TO TEST DATA.

In the application of the method to test data of a propeller use is made of the following formulae:

$$N_s = N \frac{P^{\frac{1}{2}}}{V^{\frac{5}{2}}}, \quad . \quad . \quad . \quad (1a)$$

$$D_s = D \frac{V^{\frac{1}{2}}}{P^{\frac{1}{4}}}, \quad . \quad . \quad . \quad (2a)$$

$$v_s = \frac{v}{V} \quad . \quad . \quad . \quad . \quad (3a)$$

The notation for the formulae is summarized as follows:

N_s = Specific speed, revolutions per minute.
D_s = Specific diameter, feet.
v_s = Specific tip speed, feet per minute.
N = Rotational speed, revolutions per minute.
D = Diameter, feet.
v = Tip speed, feet per minute = π DN.
P = Useful (thrust) power developed by propeller, expressed in hundreds of horsepower, and reduced to standard air density of 0.075 pound per cubic foot.
V = Velocity of forward movement of propeller relative to still air, expressed in hundreds of miles per hour.

The following data are required from a test of a propeller to which the Specific-Speed method is to be applied:

Useful horsepower reduced to standard air density.

Velocity of forward movement of propeller relative to still air.

Rotational speed.

Diameter.

Efficiency.

The computations of the specific units for a propeller from data obtained from the test of a model are given in Table I, and the method of computation is explained in detail in Appendix I.

CURVES.

The calculations of Table I yield four items which are the basis of the Specific-Speed method. These items are N_s, D_s,

TABLE I.—COMPUTATIONS FOR SPECIFIC UNITS. PROPELLER No. 1.

	Velocity, hundreds of miles per hour. V	Useful power, hundreds of horsepower. P	Revolutions per minute N	Diameter of propeller feet. D	Efficiency	ρ^4	V^4	V^5	V^6	$\frac{\rho^4}{V^5}$	$N_s = N\frac{\rho^4}{V^5}$	$\frac{V^5}{\rho^4}$	$D_s = D\frac{V^4}{\rho^4}$	v, Tip speed feet per minute $= D\pi N$	v_s, Specific tip speed $\frac{v}{\sqrt{V}}$
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1	0.2455	.02903	2,400	3.000	.483	.1704	.4955	.12166	.02985	5.7091	13701.8	.7140	2.1420	22619.5	92155
2	0.3273	.03692	2,400	3.000	.586	.1922	.5721	.18725	.06128	3.1367	7528.1	.9742	2.9226	22619.5	69116
3	0.4091	.04055	2,400	3.000	.665	.2013	.6396	.26165	.10704	1.8806	4513.4	1.2998	3.8994	22619.5	55292
4	0.4909	.04214	2,400	3.000	.722	.2053	.7006	.34388	.16885	1.2159	2918.2	1.6750	5.0250	22619.5	46077
5	0.5727	.04146	2,400	3.000	.757	.2036	.7568	.43345	.24823	0.8202	1968.5	2.1889	6.3867	22619.5	39495
6	0.6546	.03800	2,400	3.000	.766	.1949	.8091	.52967	.34662	0.5623	1349.5	2.7177	8.1531	22619.5	34557
7	0.7364	.03247	2,400	3.000	.739	.1802	.8581	.63185	.46530	0.3873	929.5	3.5064	10.5192	22619.5	30718
8	0.8182	.02333	2,400	3.000	.647	.1527	.9045	.73999	.60552	0.2522	605.3	4.8460	14.5380	22619.5	27646
9	0.9000	.00847	2,400	3.000	.412	.0920	.9487	.85386	.76843	0.1197	287.3	9.2811	27.8433	22619.5	25133

v_s, and efficiency. In order to render these items available for use in the solution of problems and in the analysis of performance they are shown in the form of curves as follows:

Curve I, Fig. 1, Specific Speed *vs.* Efficiency.

Curve II, Fig. 1, Specific Diameter *vs.* Efficiency.

Curve III, Fig. 2, Specific Speed *vs.* Specific Diameter.

Curve IV, Fig. 3, Specific Speed *vs.* Specific Tip Speed.

All of these curves are by no means necessary in the application of the method to the solution of problems, but the curves are all presented because they show some interesting and important relations between the various items. Before taking up the solution of problems some characteristics of the curves will be discussed. In analyzing the curves it is well to keep in mind the fact that, for any requirement of velocity and power the actual speed is proportional to the specific speed, the actual diameter is proportional to the specific diameter, and the actual tip speed is proportional to the specific tip speed. Thus the curves show not only the relation between the specific units for the propeller, but also show the relation between the actual speed, diameter, tip speed and efficiency of the propeller, to some scale, depending only upon the values for the velocity and thrust power of the machine to which the propeller is to be applied.

Curve I, Fig. 1, shows that a propeller of this type operates at its maximum efficiency at only one value of specific speed and, therefore, at only one value of actual speed for any given requirement of forward velocity and power. At speeds either lower or higher than this speed the efficiency falls off, but the curve is quite flat over a wide range of speeds. Curve II, Fig. 1, shows the same thing regarding the specific diameter; although there is only one point for maximum efficiency, the efficiency curve is flat over a wide range in

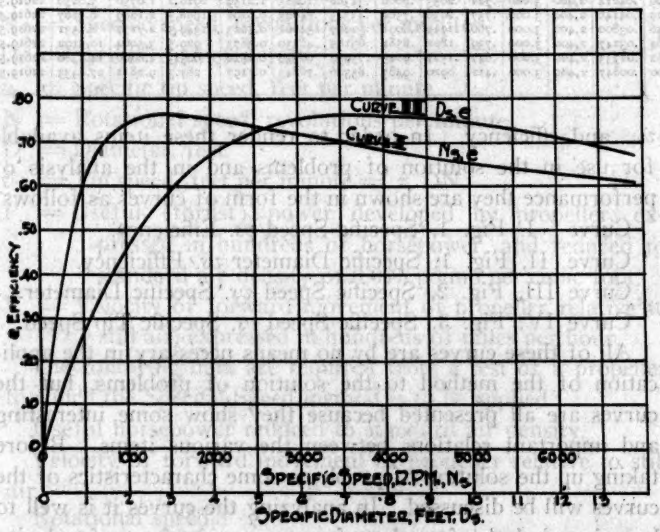

FIG. 1.—CURVES OF SPECIFIC SPEED AND SPECIFIC DIAMETER VS. EFFICIENCY.

diameters. The relation between specific speed and specific diameter is shown in curve III, Fig. 2, and it is seen that the diameter decreases with increasing speeds. The efficiencies at the various values of N_s and D_s, shown on curves I and II, are transferred to curve III, and shown as ticks along the inside of the curve. The maximum efficiency of 0.767 occurs at values of $N_s = 1{,}540$ and $D_s = 7.38$, and the efficiency decreases in both directions from this point. With the addi-

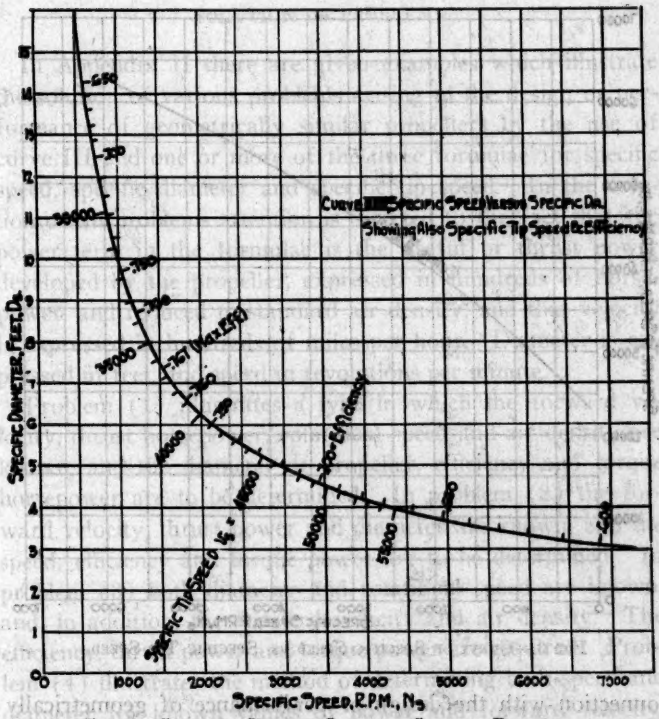

FIG. 2.—CURVE OF SPECIFIC SPEED VS. SPECIFIC DIAMETER.

tion of the efficiency ticks, curve III contains all the data shown by both curves I and II. Curve IV, Fig. 3, shows the rapid increase in specific tip speed associated with increasing specific speed. The tip speed data of this curve are transferred to curve III by picking off values of N_s corresponding to even values of v_s and plotting them as ticks along the outside of the curve of N_s *vs.* D_s. With the addition of the efficiency ticks and the tip-speed ticks curve III shows all the relations between N_s, D_s, v_s and efficiency as derived from the test of a propeller, and is the only curve required in the solution of any type of problem which may arise in

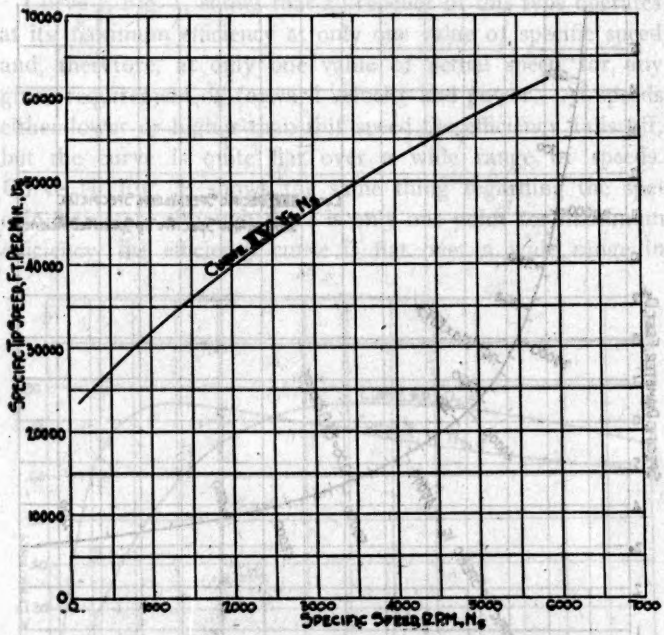

FIG. 3.—CURVE OF SPECIFIC SPEED VS. SPECIFIC TIP SPEED.

connection with the design or performance of geometrically similar propellers. The more important relations which may be observed from curve III may be summarized as follows:

(*a*) For given requirements of forward velocity and power the maximum efficiency may be obtained at only one value of N_s and one value of D_s, and therefore at only one value of actual speed and diameter.

(*b*) At rotational speeds and diameters greater or less than those corresponding to the point of maximum efficiency the efficiency decreases, but the efficiency is quite close to the maximum over a considerable range of diameters and speeds.

(*c*) As the diameter decreases both the rotational speed and the tip speed increase over the entire range of the curve.

SOLUTION OF PROBLEMS.

In Appendix II there are given examples which illustrate the solution of various problems arising in the design or performance of geometrically similar propellers by the use of curve III and one or more of the three formulae for specific speed, specific diameter and specific tip speed. In the solution of all problems attention is directed to the fact that the power term in the formulae is the useful or thrust power developed by the propeller, expressed in hundreds of horsepower, and reduced to standard air density, and that velocity is expressed in hundreds of miles per hour. Diameter is expressed in feet, and speed in revolutions per minute.

Problem (1) illustrates a type in which the forward velocity, thrust horsepower, rotational speed and air density are known, and the diameter of propeller, efficiency and torque horsepower are to be determined. In problem (2) the forward velocity, thrust power and diameter are known, and the speed, efficiency and torque power are to be determined. In problem (3) both diameter and rotational speed are known, and, in addition, the forward velocity and air density. The efficiency, thrust power and torque power are required. Problem (4) illustrates the method of determining both speed and diameter for known values of power and forward velocity. Problem (5) illustrates the method of design for known values of power and forward velocity when the tip speed is limited to a certain value. It is noted that all problems are solved in the following general manner:

(*a*) One of the specific units is determined by substitution in the formula which involves the given data.

(*b*) The remaining specific units and the efficiency are determined from the curve.

(*c*) The desired data are obtained by substitution in formulae for specific units and the usual formulae involving power, torque, thrust and efficiency.

COMPARISON OF TYPE CHARACTERISTICS AND SELECTION OF TYPE.

A further important application of the Specific-Speed method is in the comparison of type characteristics in the selection of the proper type for certain requirements. For illustration, on Figs. 4 and 5 are plotted characteristic curves for three propellers with different values of pitch ratio, but similar in other respects. These groups of curves may be used to determine which of several designs is the best suited

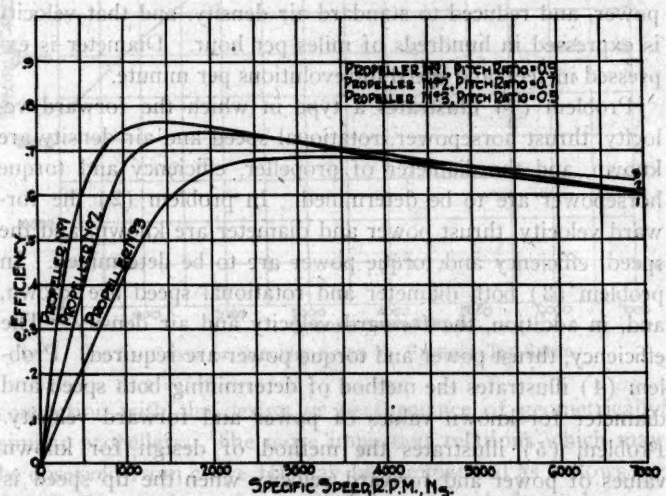

FIG. 4.—SPECIFIC SPEED VS. EFFICIENCY FOR THREE PROPELLERS.

for certain requirements. In the most usual type of problem the forward velocity, thrust power, and rotational speed are known. These determine the specific speed by substitution in the formula

$$N_s = N \frac{P^{\frac{1}{2}}}{V^{\frac{5}{2}}}.$$

The object of design is to obtain as high an efficiency and as small a diameter as possible which will meet the requirements of speed, power and velocity. Whatever propeller type

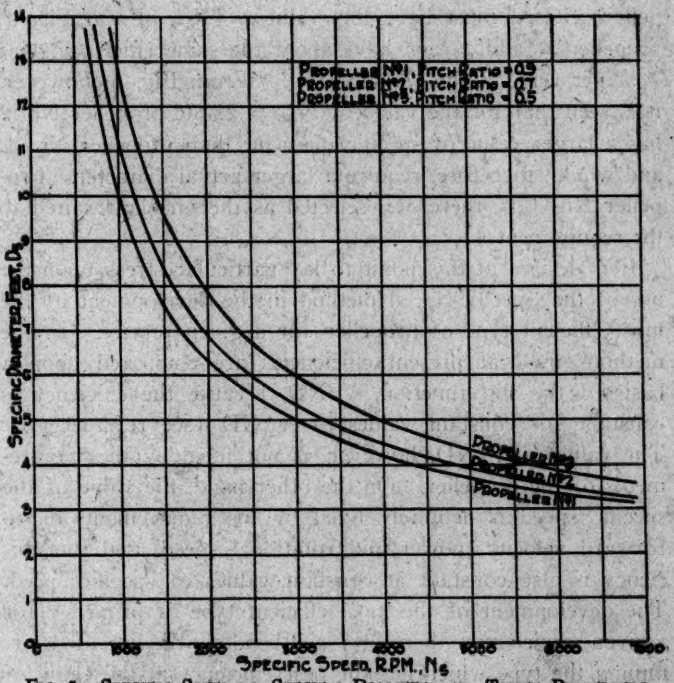
PROPELLER No1. PITCH RATIO = 0.5
PROPELLER No2. PITCH RATIO = 0.7
PROPELLER No3. PITCH RATIO = 0.9
PROPELLER No3
PROPELLER No2
PROPELLER No1
SPECIFIC DIAMETER FEET, D_s
SPECIFIC SPEED R.P.M., N_s

FIG. 5.—SPECIFIC SPEED VS. SPECIFIC DIAMETER FOR THREE PROPELLERS.

is selected and at whatever point the design is made the specific speed *must be* the value as determined by the specific speed formula above. Therefore the best type for the given requirements is determined by entering groups of curves of N_s *vs.* efficiency, and N_s *vs.* D_s, similar to those shown on Figs. 4 and 5, and selecting that type which has the most satisfactory values of efficiency and specific diameter at the value of specific speed at which the propeller must operate. For example, assume that a propeller is required which will produce 128.5 useful (thrust) horsepower and a forward velocity of 80 miles per hour, at a rotational speed of 1,300 revolutions per minute. From formula (1*a*) the value of the specific speed for these requirements is 2,580, and the pro-

peller selected must have this value of N_s. From Fig. 4, propellers 1 and 2 each have about the same efficiency, viz: 73.5 per cent, at this value of N_s. From Fig. 5, however, it is seen that for the value of $N_s = 2{,}580$, propeller No. 2 has a larger value of specific diameter than propeller No. 1, and would, therefore, require a larger actual diameter. Propeller No. 1 is, therefore, selected as the one best suited to the requirements.

It is desired at this point to lay particular stress upon the use of the Specific-Speed method in the development of the most efficient type of propeller for a given service. In the methods used at present efficiencies are compared upon a basis of the slip function, V/ND, because the efficiency is constant for constant values of V/ND (see reference *g*). The value of V/ND, however, is not fixed by the requirements of the propeller. On the other hand, the value of the specific speed is definitely fixed by the requirements as to forward velocity, power and rotational speed, and the efficiency is also constant at constant values of specific speed. The development of the most efficient type of propeller for a given service may, therefore, well be along the line of determining the type which has the best efficiency at the value of specific speed which is required for the service.

The general method as developed for the air propeller in the present paper will also apply to the water propeller, if it may be assumed that the laws of similitude upon which the method is based also apply to water propellers.

SUMMARY.

The specific speed, specific diameter and specific tip speed of any type of air propeller, operating at any given efficiency, are defined as the speed, diameter and tip speed, respectively, of a geometrically similar propeller which will develop 100 useful (thrust) horsepower when the forward velocity is 100 miles per hour. Formulae (1*a*) (2*a*) and (3) for the specific units are derived from the laws of similitude as applying to propeller performance. From test data of any propeller of

the type computations are made of the values of the specific units at various efficiencies; and the specific units and efficiencies are plotted in the form of curves, which, in connection with the formulae, cover completely the design and performance of all propellers which are geometrically similar to the propeller tested.

It is believed that the presentation of the performance of propellers and the solution of problems by the Specific-Speed method may prove to be much simpler than the present methods which involve the use of coefficients and functions for the following reasons:

(*a*) The coefficients or functions are merely abstract numbers without any tangible significance, while the specific units used in the Specific-Speed method may be thought of as speeds and diameters for unit requirements of performance, and the actual speeds and diameters for any requirements of performance are simply proportional to the specific units.

(*b*) All problems in the performance or design of a group of geometrically similar propellers may be solved by the use of a single curve and the simple formulae for the specific units.

(*c*) The characteristics of various types of propellers may be compared upon coördinates which show directly the value of the propellers in meeting any desired requirements of performance.

APPENDIX I.

Source of Items in Table I.—Computations for Specific Units.

Item.

1. **Velocity, hundreds of miles per hour = velocity, miles per hour divided by 100.**
   Obtained from test data.
2. **Useful horsepower, realized in forward movement of propeller, expressed in hundreds of horsepower = horsepower ÷ 100.**
   Obtained from test data and reduced to standard air density of 0.075 pound per cubic foot.
3. **Revolutions per minute of propeller.**
   From test data.

4. Diameter of propeller, feet.
5. Efficiency of propeller.

From test data.

6. $P^{\frac{1}{3}}$ = item (2)$^{\frac{1}{3}}$.
7. $V^{\frac{1}{3}}$ = item (1)$^{\frac{1}{3}}$.
8. $V^{\frac{2}{3}} = V^{\frac{1}{3}} \times V$ = item (7) × item (1).
9. $V^{\frac{3}{3}} = V^{\frac{2}{3}} \times V$ = item (8) × item (1).
10. $\frac{P^{\frac{1}{3}}}{V^{\frac{1}{3}}}$ = item (6) ÷ item (9).
11. $N_s = N\frac{P^{\frac{1}{3}}}{V^{\frac{1}{3}}}$ = item (3) × item (10).
12. $\frac{V^{\frac{2}{3}}}{P^{\frac{1}{3}}}$ = item (8) ÷ item (6).
13. $D_s = D\frac{V^{\frac{2}{3}}}{P^{\frac{1}{3}}}$ = item (4) × item (12).
14. v, tip speed, feet per minute, $= \pi$ DN $= \pi \times$ item (4) × item (3).
15. v_s, specific tip speed, $= \frac{v}{V}$ = item (14) ÷ item (1).

APPENDIX II.

Solution of Problems.

Problem (1).

Given: Velocity = 80 miles per hour.
Useful power = 120 horsepower.
Air density = 0.070 pound per cubic foot.
Speed = 1,300 revolutions per minute.

Determine: Diameter of propeller required.
Efficiency.
Effective horsepower.
Torque and thrust.

(*a*) Power at standard air density $= 120 \times \frac{0.075}{0.070} = 128.5$.

Then: $P = \frac{128.5}{100} = 1.285$,

$V = \frac{80}{100} = 0.80$,

$N = 1{,}300$.

(*b*) From formula (1*a*), $N_s = N\frac{P^{\frac{1}{3}}}{V^{\frac{1}{3}}} = 2{,}580$.

(*c*) From curve III, for $N_s = 2{,}580$,
we have $D_s = 5.37$ and efficiency = 0.735.

(*d*) From formula (2), $D = D_s\frac{P^{\frac{1}{3}}}{V^{\frac{1}{3}}} = 9.65$ feet.

(*e*) Effective power = 120 ÷ 0.735 = 163.5 horsepower.

(*f*) Torque and thrust are found by the usual formulae.

Problem (2).

Given: V, P, D.

Determine: N, efficiency, effective power (P_e).

(*a*) Solve for D_s from formula (2*a*).

(*b*) From curve III, at the value of D_s, determine N_s and e.

(*c*) Solve for N from formula (1).

(*d*) Determine P_s from efficiency, and P.

(*e*) Reduce power to actual air density.

(*f*) Determine torque and thrust as usual.

Problem (3).

Given: D, N, V, d. (d = air density.)

Determine: Useful power, efficiency, and effective power.

Note.—This type of problem, in which both the diameter and speed of propeller are known, is usually solved by the use of torque and thrust coefficients. It will be shown how this type of problem is solved by the specific speed method.

Given: Diameter = 8 feet.
Rotational speed = 1,200 revolutions per minute.
Forward velocity = 72 miles per hour.
Air density = 0.071 pound per cubic foot at elevation of 3,000 feet.

Determine: Thrust, torque, useful and effective horsepower, and efficiency.

For use in the specific-speed formulae,

$D = 8$

$N = 1{,}200$

$V = 72 \div 100 = 0.72$

(*a*) $v_s = \frac{\pi DN}{V} = \frac{\pi \times 8 \times 1{,}200}{0.72} = 41{,}900.$

(*b*) From curve III, for a value of $v_s = 41{,}900$,

$N_s = 2{,}240$,

$D_s = 5.8$,

Eff. = 0.746.

(*c*) From equation (2)

$$D = D_s \frac{P^{\frac{1}{2}}}{V^{\frac{1}{2}}}.$$

(*d*) Substituting values

$$8 = 5.8 \frac{P^{\frac{1}{2}}}{(0.72)^{\frac{1}{2}}}$$

$P = 0.7106.$

(*e*) Useful power at standard air density = 71.06.

(*f*) Useful power at 3,000 feet elevation $= 71.06 \times \frac{0.071}{0.075} = 67.30.$

(*g*) Effective power $= \frac{67.30}{0.746} = 90.3$ horsepower.

(*h*) Torque and thrust are determined by the usual formulae between torque, thrust and power.

Problem (4).

Given: Only P and V.

Determine: Both N and D.

From curve III, select value of N_s and D_s corresponding to best efficiency, and solve for N and D from formulae (1) and (2).

This may give values of D larger than desired and values of N smaller than desired, or vice versa. It is seen from curve III that the design may be made further along the curve, at higher speeds and smaller diameters, with not much decrease in efficiency, over quite a wide range in speeds and diameters. Also, that as the speed is increased the tip speed is increased.

A design may be selected at any point along the curve at which the efficiency and tip speed are satisfactory. The tip speed at any point along the curve is determined from equation (3*b*).

$$\text{Tip speed} = v_s V.$$

At whatever point the design is made, simultaneous values of N_s and D_s, as determined from the curve, must be used.

Problem (5).

Given: P, V, and v.

Determine: N and D.

(*a*) From equation (3*a*) determine v_s.

(*b*) From curve III, at this value of v_s, determine values of N_s, D_s, and e.

(*c*) Solve for N and D from equations (1) and (*e*).

(*d*) Determine effective power, torque and thrust by the usual formulae.

APPENDIX III.

Bibliographical Notes.

The first application of the Specific-Speed method to rotating machinery was in connection with hydraulic turbines, where the specific speed was defined as the speed of a geometrically similar turbine which was of such size that it would develop one horsepower under one foot head (see reference c). The application of the method to the centrifugal pump was proposed by Greene (reference d). Here the specific speed of the centrifugal pump was defined as the speed of a geometrically similar pump which would pump one unit volume of water against a unit head. The application of the method to the centrifugal fan was made by the writer (reference e), and the method was extended to include the units specific diameter and specific tip speed, formulae for which were developed. In the early use of the Specific-Speed method it was not realized by all that the specific speed of a given type of machine varied over such wide limits with efficiency, and the specific speed of a machine was taken as the specific speed at the point of best efficiency. Owing to limitations of diameter or speed it is not always possible to design at the point of best efficiency, and the design must be made at some other speed and diameter at which the efficiency is slightly less than the maximum. From this it

is seen that it is not sufficient to simply refer to the specific speed of a machine as a definite value, but the relation between specific speed, efficiency, and specific diameter must be considered. Application of the method to the centrifugal pump was treated in detail by Hechler (reference f).

The extent to which results of experiments made on model air propellers may be applied to full-size propellers is discussed in reference (b), page 597.

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(a) Report No. 14, "Experimental Research on Air Propellers," National Advisory Committee for Aeronautics, 1917.

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(c) "Characteristics of Modern Hydraulic Turbines," Chester W. Larner, A. S. C. E. Transactions, 1910, page 306.

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(f) "Characteristics and Specific Speeds of Centrifugal Pumps," F. G. Hechler, A. S. N. E. JOURNAL, November, 1917.

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SCREW PROPELLER MANUFACTURE.

By Commander U. T. Holmes, U..S. Navy, Member.

The following notes on propeller manufacture have been prepared from observations of the methods employed at one of our leading shipyards in *manufacturing* screw propellers. The importance of having propellers strictly in accordance with designed dimensions has been made evident in recent years, during which comprehensive studies have been made, with very great improvement leading almost to perfection.

Given the form, displacement and speed, with number of shafts and speed of rotation of the shafts of a vessel, it is now possible to fix upon the details of the screw that will transmit the necessary effective horsepower with the least expenditure of shaft horsepower. The first element in the problem is to determine the effective horsepower required. This may be obtained from towing experiments in the model tank.

Extensive researches along these lines have been carried on, going back over a period of at least fifteen years, a critical time in the development of high-speed, high-powered vessels. The work of two officers in our Navy has been especially noteworthy in this connection. Rear Admiral C. W. Dyson has during this time been untiring in the work of investigating propeller data. He has been able to perfect his method of designing propellers by having available the coefficients obtained in towing experiments with models of many of the vessels under investigation. In this he was assisted in a notable manner by the work of the present Chief Constructor of the Navy, Rear Admiral D. W. Taylor, in directing the experimental work at the Naval Experiment Tank, Washington Navy Yard. If provided with the model-tank data it is

now possible to fix upon the most suitable propeller when working out details of the machinery of any given vessel.

CHARACTERISTICS OF RECENT PROPELLERS.

Recent designs for high speed require propellers to be true screws, with solid hubs, and usually with three blades. The hub diameter is usually about one-sixth of the propeller diameter. Backs of blades are given a contour such as will afford the necessary strength, with least weight, while minimizing eddy-making resistance. Hubs are fitted with conical-shaped caps the fore part of which continues the contour of the after part of the hub. Frequently the cap is also the propeller nut.

CASTINGS.

At these works, when a large number of propellers are to be manufactured of the same design, special facilities are provided to turn out the castings expeditiously. Details of the foundry work are as usual in loam work. The following points in connection therewith are of special interest:

Hubs are cast with a small core to facilitate boring out. The axis is vertical in casting with the gate for inflow of metal at the bottom of the hub and a very heavy riser over the hub. An overflow gate is placed at the upper edge of each blade to ensure the metal filling the mold completely. A test piece is provided for cutting off of the hub and off of the face of each blade.

Backs of blades are cast downward and are molded as nearly true to form as is possible. About 5/8 inch excess thickness is allowed on the faces of blades, with a margin of approximately 2 inches in width around the edges. In the castings the edges are not fined to less than about 1 inch in thickness. This excess metal in the castings is necessary to guard against the inevitable warping that occurs when they cool, and assure that they will machine out true to form.

The heads, risers, gate pieces and test pieces are removed in foundry before sending to shop. Tensile-test pieces are pre-

pared and pulled in the foundry laboratory, where the testing machine is located.

MACHINE WORK.

Before any work is done on a propeller casting it is carefully measured and centers are located for boring out and turning off the blade tips. It is then centered on the table of a vertical boring mill, the hub is bored out and faced to finish size, and tips of blades are cut to the designed diameter.

Temporary keyways are now cut for holding on mandrel while finish facing blades. These are in the position of the permanent keyways, but with diminished dimensions.

The machines for facing blades take the casting on a horizontal tapered mandrel, specially made for each job in hand, with taper conforming to that on the end of propeller shaft.

Where a considerable number of propellers are in hand, made to the same design, the finish keyways are cut before machining faces of blades. The mandrels used in the machines are then made with keys conforming to those used on propeller shafts.

The cutting tool is mounted on a tool post carried by a feed screw along the slide ways of a specially constructed machine. This machine in operation causes the tool to advance while the mandrel rotates, the relative speed of cutting tool and propeller being adjusted by change gears so as to cut to the pitch required. The motion is reversed automatically, as required, by adjustable stops on the bed of the machine. The cutting tool is advanced before beginning the advance movement by an attendant who adjusts the feed to suit the work and guards against casualties in operation. In later machines the cutting tool is advanced automatically.

Rough cuts are taken over the face of each blade until all are brought to within about 1/32 inch of the finished surface. A finishing tool is employed, with the same setting, for finish cuts on all blades.

After completion of this work the propeller is placed, face

up, on a laying-off platform; the faces are whitened and center lines of blades are drawn to aid in drawing contour lines all around. These are center punched lightly at intervals to prevent obliteration. The center of blade at the end is prick punched for transfer of the center line to the back. The propeller is then turned over, the center line is transferred to the back, lines are drawn at regular intervals parallel to center line; also concentric curved lines at regular intervals measured from the propeller center. All these lines are painted on with marking ink.

The next step is to caliper the thickness of blade at each intersection, then the difference between this and the known finished thickness is painted on, indicating the amount of metal that is to be taken off.

The work is now ready for the finishing gang. Their first step is to chisel off the excess width around edges, using pneumatic hammers. The same process is then applied to the whole back, where necessary, first cutting grooves about ¼ inch wide along the various black lines to a depth nearly approaching that indicated. Flat chisels are then used to bring the whole surface down flush with the bottom of these grooves. The final finish on back is with heavy rough files and portable grinders.

The fillets against the hubs on faces are rounded out in much the same manner as is employed in finishing the backs. The faces are finished all over by scraping where necessary and polishing with blocks faced with emery cloth.

When the finish work is nearly completed a mandrel is fitted and the propeller is mounted on horses for final finish and balancing. When completed the wheel is required to balance exactly in any position.

All that remains after balancing is the cutting of keyways and the fitting of caps and glands. This work requires no special mention except that in these works the finished bore, with keyways complete, must be made to fit exactly on a cast-

iron mandrel with keys fitted, that exactly represents the end of the propeller shaft.

Finally, with the manufacture completed, the hub of the finished propeller is stamped with the mark showing that it has passed inspection and also with identification marks to individualize it. These include, beside the inspector's special stamp, the heat number, the weight, the name of the vessel, if for a particular vessel, initials indicating right or left hand, and any other special marking that may be directed.

BENJAMIN F. ISHERWOOD.

NOTES.

A. S. M. E. PRESENTED WITH BUST OF ADMIRAL ISHERWOOD.

At its annual meeting in December, 1917, the American Society of Mechanical Engineers was presented with a portrait bust of Admiral Benjamin Franklin Isherwood, who was for many years, up to the time of his death, an honorary member of the society. The bust was a gift from a number of the friends and admirers of the Admiral. The presentation address was to have been made by Commodore George W. Magee, U. S. Navy, who had been an old friend and assistant of Admiral Isherwood; unfortunately, however, Commodore Magee was taken ill, and that pleasant duty devolved upon W. M. McFarland.

Mr. McFarland pointed out that the special glory of Admiral Isherwood's work was that he helped to establish a number of the important basic principles of the science of engineering. He stated that the first reproduction of actual indicator cards from a marine engine published in any book was in Isherwood's work on "Engineering Precedents," published about 1856. He said further:

"His reports of experiments are models of what such reports should be. They include a complete description of the apparatus; and the log of the experiment, in each case, gives all the data which could be observed, whether they were immediately applicable to the purpose of the experiment or not. The result of this is that these reports constitute a mine of valuable information; and other engineers, many years after, seeking information on an entirely different line from that for which the experiment was conducted, find in the complete and careful record just the information they want, and which often can be found nowhere else."

The written address of Commodore Magee, which was read by Mr. McFarland, referred to the great value of Isherwood's experimental work on such subjects as the expansive working of steam, screw propellers, the use of superheated steam and the use of forced draft. Commodore Magee characterized Admiral Isherwood as "the greatest marine engineer who has thus far appeared in our country," and "not only a great engineer, but a great administrator and executive."

Dr. Hollis, President of the A. S. M. E., in accepting the bust on behalf of the society, spoke of his personal acquaintance with Admiral Isherwood at the Naval Academy and said he considered his service for the Admiral as perhaps the proudest of his life. Continuing, he said:

"I think of him as perhaps the father of our great research laboratories in engineering, as his investigations in connection with steam engines and with boilers preceded all of our schools of mechanical engineering. I think of him also as the father of high speed on the sea. Few people realize that it was Benjamin F. Isherwood who during the Civil War planned and carried to its completion the first ocean greyhound, a ship which went outside along the Jersey coast and made four hundred miles in one day with ease, something that was not equaled again for twenty years."—"Power."

MELTING METALS.

By Walter J. May.

In recent times there appears to have arisen an idea that because a metal requires so many degrees of heat to ensure that it is in a state of fusion, and that as a certain weight of fuel—usually metallurgical coke—contains the possibility of producing that number of degrees of heat under suitable conditions in regard to combustion, all that has to be done is to burn the fuel in more or less intimate connection with the metal and the metal melts. No attention is paid to the condition of the metal, the form of furnace, the time required by the metal for the absorption of heat, or other practical details, and above all no consideration is paid to the condition of the molten metal in regard to the purpose to which it is to be placed. Of course, up to a certain point the heat content (when developed) of the fuel has to be considered, but merely because Professor X states that 3 pounds of fuel has a sufficient thermal content to fuse 100 pounds of gray iron, it does not follow that in practice 3 pounds of fuel will reduce that weight of iron to a condition fit for making castings, while probably if the total heat evolved was suddenly put on 100 pounds of pig iron the only effect would be a slight warming of the outside of the pieces of metal. Not one, but all conditions used have to be considered, and in practice there has also to be taken into very serious consideration heat losses which are unavoidable, and under the best conditions and care in working, Professor X's 3 pounds of fuel will come out at from 9 pounds to 12 pounds. This does not for one moment imply that Professor X is wrong, but it points out that his conditions cannot be reached in practical working, and it is here that a great amount of wasted experiment is made in trying to secure that which is unattainable outside the laboratory. What *should* be and what is actually secured differ very widely, and when we consider that the primary purpose of all foundry melting is to secure molten metal fit for the production of castings, the question of fuel consumption outside certain fairly well-defined limits becomes a minor matter when compared with some other points in regard to the production of the metal, sound castings at a remunerative total cost being the chief object in view.

As metals at certain crucial temperatures will absorb matter in a gaseous state, it follows that fuel should be as pure carbon as possible, sulphur and metallic salts being as far as possible absent. Usually, gas coke is dirty, some samples being worse than others, and for this reason clean and specially prepared metallurgical coke should be used. The cost per cwt. of metal melted is less with the metallurgical coke, and the plumbago crucibles last longer, and really, although the cost is about double that of gas coke where mere tonnage is concerned, the metallurgical coke costs in the end much less, as a smaller weight of fuel and fewer crucibles are needed to melt a given weight of metal.

Time is an important point in metal melting, because the absorption of heat is gradual, varying according to the bulk of the pieces of metal dealt with. Fairly light scrap comes down quicker than ingots, and ingots put into a bath of molten metal will absorb heat quicker and fuse sooner than those stood in an otherwise empty crucible. Wire gauze and the like will only melt in a bath of metal, such things simply burning away when otherwise dealt with, and in some furnaces such things as fine turnings and borings can only be reduced in a bath of metal. Aluminum scrap has to be treated with other than mere heat when light stuff is being handled, and zinc scrap will often need special treatment to get good returns in metal, although solid metal presents no difficulty in melting. These details, although well known to the actual melters of metals, are often too much overlooked by those who have charge of the foundry department, there

usually being too much passing over of what really is of importance, and this in itself causes an appearance of waste—or economy in some cases—which has no real base in practical working. Aluminum is a disconcerting metal to melt when one is not up to its little vagaries, because while taking only a low heat to melt it fit for pouring, the bars of metal will often stand in the crucible a long time before they drop in a liquefied state, and then they collapse suddenly. On the other hand, if we place a bar in a bath of molten brass we wait a very short time and then there is a flash of brilliant flame, at times a small explosion of little importance, and then the alloy of brass and aluminum is complete, a bar which melted by itself would take quite half an hour to bring down being alloyed with a pot of brass in some five minutes or so. In making copper-tin alloys also, if we first melt our copper and then add the tin we have to burn much more coke than if we bring the copper to a full heat and then place the tin on top of it, as in running down the tin brings down the copper, provided the mass of the copper is already heated to within some 500 degrees F. of fusing point, the content of the copper, of course, exercising some influence. In nearly all cases where melting or alloying is done, granulated charcoal should be in the crucibles, but in both air furnaces and cupolas the reducing atmosphere has to be maintained by other means.

Where, however, the freedom from carbon is a necessity, as, for instance, in Mitis iron, bronze containing a high percentage of iron, and a few other things, charcoal cannot be used in the crucibles, close covering being practically all that can be done; but as little free oxygen as possible should be present in the furnace, a very careful balancing of the forced draft with the fuel consumption being needed, and no hard-and-fast rules can be laid down in the matter. Certainly, there is no great difficulty in alloying carbon-free iron and copper together in almost any proportions, but when carbon gets into the metal trouble arises. Bronze made of copper, iron and zinc chiefly, is a practically incorrodible alloy, but necessarily there must be a limit to the percentage of iron for physical reasons.

In arranging for melting, metals and alloys should be largely grouped according to their melting points, and this will permit the furnace man to arrange his fires according to the heat which has to be used to bring the metal to its most fluid state for pouring, while incidentally a considerable economy in the cost of crucibles can be effected in the process, a less refractory material being needed for soft alloys than for iron and copper. In some cases the crucibles need liners for dealing with particular metals and alloys, but this does not affect the refractory state or grade of the crucible proper.

In the actual melting it is really important that the crucibles should be closely filled to ensure that the heat transmitted through the crucible walls shall as far as possible be picked up and absorbed by the metal, it being advantageous also that covers be on the crucible to retain the heat, although these need not necessarily be luted on in many cases. It is also necessary that the fuel used should be of moderate size, free from fine stuff and dust, and that the furnace be properly packed with fuel to ensure the maximum heat for the period necessary to melt the metal, due consideration being paid to the draft available in each particular furnace.

With cupola melting it is necessary as far as practicable to prevent the heat creeping up the furnace walls and escaping without effecting its proper work, and in this respect a large amount of care is necessary in putting on the charges. Indeed, the man who is entrusted with the care of a cupola requires to have a goodly amount of knowledge of the behavior of his furnace and how to get the greatest melting effort from the material used, a close, sticky mass in the center with a free escape of heated gases around the walls of the furnace not conducing to the best

results in either melting or quality of metal. It is really quite possible to fool away from 25 to 30 per cent of the weight of metal in fuel where 12 per cent should be more than ample to secure far better results, but in all cases the matter is largely a personal one, although the general arrangement of the furnace and blowing apparatus also has much to do with the matter.

In the air furnace the general arrangement and the fuel used usually govern the results obtained, although the placing of the metal to be melted into particular positions will considerably alter the rate of melting. Individual practice in charging causes many variations in the results obtained, and for these reasons one man should always be placed in charge of the furnace.

With forced-draft crucible and other furnaces there is less left to clearance than with those having natural draft only, provided the man in charge has a good knowledge of melting and the relation of the proportion of air to fuel, the working being more of a positive than permissive state of things. The furnaces for forced-blast work are designed to give the best results under certain clearly defined conditions, to depart from which means some larger or smaller amount of inefficiency, for which reason the operator *must* follow the instructions given by the makers of the particular apparatus which is being used.

Gas furnaces may very well be used for some metals, and the same remark applies to oil furnaces; but unless care is taken the crucibles get badly cut, and the results do not prove quite so satisfactory as one is led to expect when the total cost of the production of castings is considered. Really, either gas or oil should give better net results than those obtained from solid fuel, but for some reason this does not appear to be the universal experience of users of such furnaces, and possibly something must be allowed for insufficient knowledge of manipulation, plus individual personal prejudices against innovations which by a certain class of workers are resented. There is too much of the "My grandfather always carried a stable lantern when he wanted light on the road, therefore why should we put up gas lamps? What was good enough for him should be good enough for me," sort of feeling still existent, and this largely prevents progress, although there are also defects in many of the furnaces which should be overcome by the makers.

Beyond the mere reduction of the solid metal to a condition enabling it to be poured into a mold quite independently of the kind of casting that will be produced, there is the further necessity that the metal shall be clean and produce sound castings, this causing some form of other than heat treatment to be necessary. As a general thing the removal of occluded oxides and the solution of the pasty metal with which the particles of oxide are surrounded forms the objective of the treatment given, it being usual to use some chloride or fluoride—with possibly the addition of some other agent or agents—for this purpose, but in the case of many metals a chloride proves most effective. Salammoniac and zinc chloride as a dry salt are two commonly used chlorides for non-ferrous metals, and for iron a more or less properly prepared iron chloride, or an alloy of aluminum, will be found in general use. For specific purposes other salts and alloys are also used with more or less successful results, the object with which these substances are applied usually being well defined. Mostly, however, "fluxes" act simply as deoxydants and nothing more, but some of the proprietary articles are most wonderfully concocted, most of their constituents either being inert or counteracting one another's action. As a rule a simple chemical will effect all that is needed so far as the production of clean metal is concerned, the cost being extremely small in ordinary times. It may be particularly mentioned, however, that there is no one so-called

"flux" that is effective with all metals alike, and even boron, which is effective with copper and copper alloys, in one or other of its commercial forms, will form bad compounds with some other metals and alloys; therefore each "flux" must be carefully selected for the metal and purpose for which it is intended to be used.

Apart from all mechanical considerations there is a great deal in the human element part of the matter, some men being able to secure high results apparently automatically, whilst others can be taught and drilled for years and then produce only partially successful results. It is impossible to knock into the heads of some men that iron requires a greater heat than brass or aluminum, while they will burn lead into thick porridge and send zinc up the chimney. Also one cannot prevent some men from stewing molten metal for hours "because it might be wanted today," and such men waste crucibles, fuel and metal quite foolishly, because the longer molten metal stands the worse it becomes for making castings. Metal of all kinds must be poured as soon as ready if the best castings are required, and the pouring heat is not much above the melting point, although it varies in different metals and alloys; but in any case the molds should be ready before the metal is fit to pour. Surplus metal, or that which cannot be at once used, should be poured into bars or ingot, for remelting instead of being kept in a molten state, keeping metal melted for long periods causing it a lot of injury as well as altering the content of alloys in many cases.

Taken as a whole, melting is as important as molding in a foundry, and deserves the same attention, and in its own way, skill, the employment of any spare laborer for the job irrespective of his training being rather an ill-advised proceeding, although often adopted. To melt properly and well a man must have a lot of practical training, or there will be a good many castings returned for metal faults.—"Mechanical World."

TUNGSTEN AND HIGH-SPEED STEEL.

The purpose of the present article is to describe the processes of making high-speed steel by alloying pure tungsten obtained from wolframite with iron. The former metal is used to the extent of 18 per cent to 20 per cent, and the essential and remarkable qualities of the alloy are mainly due to its use.

Tungsten powder is, of course, also produced abroad, but British requirements are now supplied from Widnes. No other alloy equals the high-speed steels and vanadium high-speed steels manufactured by makers of repute in Sheffield and Manchester; they are undoubtedly superior for all purposes where heavy and continuous work is required. The processes we describe are essentially those employed by the companies obtaining their supplies of tungsten from the Widnes works; in the aggregate these firms form a very important portion of industrial Sheffield, both as regards output, capital and as employers of labor. These companies have not only maintained a world-renowned reputation for the production of tool steel, they have also given a lead to the engineers of the world by the splendid results obtained by means of their high-speed steels, and more especially by the more recent introduction of tungsten-chrome-vanadium high-speed steel. The consumption of tool steel is very great at the present time; the companies referred to above had a combined output of high-speed steel prior to the war of from 70 per cent to 80 per cent of the total output of the country.

High-speed steel obtained its name on account of its qualities and characteristics, which may be summarized as follows: the great increase

in the surface depth of cut and speed which can be obtained; and the high speed with which work can be turned out of the shops. The increase in output, however, is not only due to depth of cut and surface speed, but mainly to the quality of retaining a cutting edge or face sufficiently hard to resist the wear and pressure of the chip being removed from the stock, even when the tool nose becomes red hot, which gives high-speed steel its great advantage over carbon steel. It may be interesting at this point to give a brief résumé of the various processes of manufacture, which are as follow:

1. By the well-known melting hole or furnace, in which the material is melted in fireclay or plumbago crucibles, the heat being obtained from coke by natural draft.
2. The same method as the first, but with producer gas in place of coke.
3. By means of the Siemens furnace.
4. By the same process as the first, but with the addition of forced draft in place of the natural draft.
5. By melting in one of the varieties of the electric furnace.

The first method of melting steel was introduced into Sheffield as far back as 1740 by Huntsman; it is by his method that the town has built up its reputation for tool steel, which it enjoys at the present time. Methods 2, 3, 4 and 5 have advantages, and several makers claim special virtues for their particular output, nevertheless the crucible process (methods 1 and 2 above), in spite of its considerable antiquity, continues to maintain its superiority over the other processes; it is therefore the method of manufacture preferred, practically without exception, by all companies of repute. The plant required for the first process is as follows:

(*a*) The melting furnaces, which consist of a series of small chambers called "holes," built below the floor level of the furnace room, the top being closed by a firebrick cover, whilst the bottom is built around a cast-iron frame, which supports a series of fire or grate bars. There is a flue from the top of each "hole" connected with the chimney stack, by means of which the draft necessary for the melting is obtained. These furnaces are each large enough to accommodate two crucibles or pots and the necessary fuel (*i.e.*, coke) for melting.

(*b*) The crucible or pot and its lid.

(*c*) An annealing grate in which the fireclay crucibles are preheated or annealed before being placed in the melting furnace.

(*d*) High-grade coke with a sulphur content as low as commercially possible.

(*e*) Cast-iron molds of various shapes into which the metal is poured when melted to form ingots.

(*f*) "Dozzles" or fireclay "feeders," which are put on the top of the ingot immediately it is cast, to reduce the amount of "pipe," or, in other words, the unsound and therefore unusable portion of the ingot, to a minimum.

The essential materials, apart from tungsten, required for the production of high-speed steel vary to some extent in the different works engaged in the manufacture of this class of steel. In most houses of repute the basis is the exceptionally pure form of iron imported from Sweden and sent to this country in the form of iron bars, hammered or rolled, according to the quality of the steel to be made. If the iron has been previously "cemented" it is easily broken to the required size.

The standard section is 3 inches by $\frac{5}{8}$ inch, the lengths varying as may be found convenient. The first step in the process is to cut up these bars into small pieces, square or triangular, of, say 2 square inches or 3 square inches of area each; these, coming straight from the store, may be more or less covered with rust, except on the cut surfaces; most of the firms

use them in this state, but some remove the rust so as to render the final product as free from the oxide as possible. This is effected by throwing the material into a rapidly revolving "rumble," in which the rust, or oxide, is effectually removed, but this process is not absolutely essential. In addition to the Swedish iron a proportion of suitable scrap is also used; a considerable supply obtained from various sources is available; its use reduces the consumption of imported Swedish iron and constitutes an important and desirable economy.

The second step in the process of making high-speed steel is a very important one. According to the class of steel required the ingredients have to be mixed with the greatest care and accuracy in accordance with detailed specifications, in which the weights and quantities have been worked out. The department in which this operation takes place is known as the "weigh-up house," and in some of the works as the "medicine house." The materials as weighed are put into a flat, open pan, a separate weigh pan being reserved for each pot or crucible. The materials mixed and weighed are the Swedish iron, scrap, tungsten, chromium and, when necessary, vanadium, molybdenum, manganese, silicon and carbon. Some manufacturers substitute other alloys for one or more of the foregoing, but they need not be further referred to in this article, which is limited to Sheffield's best quality of high-speed steel. The weigh pans, with their carefully sorted and weighed contents, are transferred to the adjoining melting house, where the crucibles are being raised to such a temperature as to make it safe to transfer the contents of the weigh pans to them.

The melting house is a building containing two ranges of furnaces, one down each side. These furnaces are, as stated above, a series of small chambers built below the floor level, with movable covers level with the floor and vertical flues built in the thickness of the walls.

Two "pot stands"—which are generally the two halves of an ordinary firebrick "square," but in some cases are of special preparations—are placed on the grate bars of the furnaces or holes, one for each of the two crucibles in each furnace. The crucibles, which are about 18 inches in height and 9 inches in their greatest diameter, are very rapidly transferred from the crucible annealing grate and placed one upon each of the pot stands already in the furnaces. Live coals are heaped round the bottom of the crucibles, and the fire is built up with coke or coal, as the melter may consider desirable. After the lapse of possibly an hour, or an hour and a half, the temperature is such that it is safe to introduce the materials to be melted into the crucibles. The contents of a weigh pan are emptied into each crucible by means of a metal funnel or cylinder known as a "charger," about 4 feet in length, with the opening wider at the top than at the bottom. When the charger mentioned above is in position a man with a long poker or "potter," as it is called, prevents the fall of any specially large piece from the weigh pan by breaking it, and thus avoids any possible damage to the crucible. This process is the "steeling of the pots." Steeling completed, the lids are carefully placed on the crucibles, the fires built up with coke until the crucibles are completely covered, and the melting process commences. The charge is left in the crucible for approximately 4 or 4½ hours, when the metal, then in the form of high-speed steel, but in a molten condition, is ready for "teeming." Ingots of almost any weight can be cast, from 40 pounds to 10 tons or 20 tons, where the united capacity of the furnaces will allow that to be done.

The weight melted in each pot is subject to variation in different works, but on an average is approximately 60 pounds per pot. To each of the men in a gang some special duty is allotted. There are two "pullers" for every 24 crucibles, and their duties are to pull the crucibles out of

the melting holes on to floor level. Having lifted the crucible out just clear of the top of the hole, they slowly revolve the pot whilst one of the other men chips away the slag and dirt which have adhered to the bottom of the crucible. This is done for the double reason of not only reducing the weight that the teemer has to lift, but also to make it easier to put the crucible, when empty, back into its place in the hole. When the slag has been removed the puller slides the crucible along the floor to the place where the teemer is waiting to do his share of the work.

Molds.—Alongside is a series of cast-iron molds, which are generally about 3 inches square in cross-section, and 30 inches in length. They are arranged so that the top is at a convenient height for the teemer, who, by means of a special pair of tongs, has to pick up the crucible and slowly pour the white-hot contents into the mold. To do this properly requires considerable knack and skill, because it is essential that the stream of molten metal shall from the first go straight to the bottom of the mold without even momentarily touching any of the four sides during the operation. When the metal has risen to within, say, 3 inches of the top of the mold, a "dozzle," which has been previously heated in one of the holes, is quickly dropped on top of the metal in the mold, and held in position whilst the few pounds of metal remaining in the crucible are poured through its center.

Dozzle.—The dozzle is a small block formed of fireclay, just large enough to fit into the top of the mold, about 4 inches long, with a hole from 1¼ inches to 1½ inches in diameter running longitudinally through its center. It serves to keep this last portion of the metal hot, and as the ingot cools, and also contracts, the portion in the dozzle tends to sink down or feed into the main body of the ingot and so fill what would otherwise be a cavity or pipe caused by the shrinkage of the cooling metal. The dozzle not only reduces the pipe in the top part of the ingot, which at all costs must be discarded, but also minimizes the weight of metal immediately surrounding the part, which of necessity has to be scrapped.

Topping.—The process of eliminating the pipe is called "topping," and consists of breaking pieces from the top or dozzle-feed end, of the ingot till all "pipe" or unsoundness has been removed and only good, sound steel remains. This topping also serves to reveal any fracture of the ingot, and from this the experienced eye is able to judge whether or not the material has been properly treated during the melting process.

The empty crucibles are then lowered into the furnace to be reheated, and as soon as the necessary temperature has been reached they are refilled with another charge; the operations described in detail above are then in due course repeated.

Analysis.—Before the ingots are available for other processes they are kept in hand until the chemist has made a complete analysis. A typical analysis of the tungsten-chrome-vanadium variety of tool steel, which, as already stated, has a leading place amongst the competing steel alloys where continuous hard work has to be faced, is approximately as follows:

	Per cent.
Carbon	0.65
Managanese	0.20
Silicon	0.20
Phosphorus	0.010
Sulphur	0.075
Tungsten	18.5
Chromium	3.75
Vanadium	0.85

Having ascertained that—

1. The melting process has been properly conducted;
2. The analysis is within reasonable limits of the specification;
3. The surface of the ingot is smooth and clean;

the ingot is allowed to go to the forge.

Forging.—The ingot is first put into a furnace in which the heating is very slow and gradual from almost cold to a dull red. It is then transferred to a hotter furnace, and is kept there till thoroughly "soaked." If the ingots have been carefully and thoroughly soaked the forging of the steel presents no very serious difficulties, but should any hammer work be attempted before the heating has been done it would almost certainly result in the ingot breaking up in all directions, or at any rate cracking in such a way as to render the steel useless for the manufacture of cutting tools. The reduction in size by the hammer is at this stage very slight, and is called the "cogging process." After cogging the bars are allowed to cool, being placed in closed boxes or in hot ashes so as to avoid too rapid cooling.

Pickling.—The bars are next pickled in a bath of hot dilute sulphuric acid or some suitable substitute, such as niter cake. The acid removes the scale or oxide that has formed during the cogging process. The bars, on being withdrawn from the acid, are carefully washed in clean water, or in some works in lime water. In this state any slight cracks or other surface defects are readily detected, and the bars are looked over before any rust begins to form. The duty of the "looker over" is to detect even the slightest hair-like crack or other defect, and near these he paints a white mark. The bars so marked are placed on special benches, where the defects are carefully ground away by means of corundum wheels. The ground bars are now ready for the finishing process.

Finishing.—Finishing is done by one of three means: (1) The forging hammers (any hammer with a capacity of over 12 cwt.); (2) the "tilts" (small hammers of under 12 cwt. capacity); or (3) the rolling mill. The particular process to be used depends upon the required size of the finished bar, and, as a more or less general rule, larger sizes are hammered, medium sizes tilted, and small sizes rolled. In any of these methods the temperature at which the material is worked is of the greatest importance, just as in the case of cogging. When a sufficiently high temperature is maintained no serious difficulties are met, but any attempt to work at a low temperature invariably brings disaster and the bar becomes scrap.

Tilting.—In the tilting process the bars are gradually worked to the desired section and size under small power-driven (steam or pneumatic) hammers of from, say, 4 cwt. to 12 cwt. capacity. These hammers do their work by a series of very rapid blows, which may be heavy or light, as desired. When the required dimensions are attained the bar is straightened and allowed to cool, having first been protected from cold air, as in the case of the cogged bar.

Rolling.—In the alternative rolling process the bar is rolled, generally in the old fashioned "two-high" mill.

Annealing.—There are numerous types of annealing furnaces, but the general principle of each is the same. In these furnaces several bars are generally packed together in a more or less cold furnace. The temperature is gradually raised, and having reached the desired maximum the heat is maintained at this for about an hour, and the furnace and contents are allowed to cool as slowly as possible, after which the bars are again pickled.

Inspection.—Finally the bars are taken to the steel warehouse. Here they are most carefully looked over for surface defects, and a small piece

is broken from both ends of each bar. Those showing the least defect are discarded or scrapped, whilst the sound ones, after being marked with the maker's name, are ready for delivery to the manufacturer of the tools for which the steel is intended.

The Manufacture of the Crucibles.—It is all-important for the successful production of high-speed steel that the crucibles should be perfect, and the greatest care is devoted to their production, as even the slightest defect might result in the loss of the metal when they are charged. These crucibles are generally made at each of the works producing high-speed steel, and a department with its special staff of trained men is set apart for the purpose. A variety of carefully selected clays are used, very finely ground, which are mixed as regards quantity and quality in accordance with the judgment of the pot maker in charge of the department. The clay is carefully weighed out in suitable quantities, then spread on an iron floor, known as the "treading" floor, which is a large rectangular tray with sides 6 inches in height, on which it is thoroughly sieved and mixed in the dry state, and any foreign matter found is immediately removed. Water is then added, and the clay is again thoroughly mixed by means of a shovel, kneaded and compressed to force out air, also for the detection and removal of any substance such as a pin, piece of wire, or anything else which might otherwise get into the substance of the crucible. This is the so-called treading process, which has been compared to the kneading of bread, but instead of being done by the hands, as in the case of bread, it is done by the naked foot. The foot being sensitive, any objectionable or dangerous foreign matter which may have slipped through the sieve by accident can be detected and removed. The clay is then cut into blocks and weighed, each sufficient for the production of one crucible, and taken to the "balling table" to be "balled"; that is, beaten, tossed and banged by hand, and again carefully examined for air pockets or cavities. The experienced eye of the "baller" can at once detect the presence of any air pockets by the slight swelling on the surface of the clay produced as the mass is banged on the balling table. Any such swelling is opened out and the air liberated; the process is continued until the block of clay has been brought to a thoroughly homogeneous mass. The blocks, about 12 inches by 9 inches by 6 inches, are then ready to be transferred to the "plug and flask," in which the plastic clay is plugged or molded to shape by means of a ram forced vertically into a mold. The pots are then removed to a drying house, or in some works to a cellar, where they are kept for at least a month, during which the moisture gradually dries out. They are then transferred to the furnace house described above and placed on shelves fixed to the walls above the furnaces or melting holes until they are required. About 21 hours before that the crucibles required for one day's work are placed in an annealing grate built at one end of the furnace house and are placed on bars with just enough space between the crucibles to allow them to be completely packed and covered with small coke until the grate is filled to the top. Sufficient fire is then put underneath the grate bars to start the coke burning at the back and bottom of the annealing grate. The fire gradually works its way forwards and upwards, until the whole mass of coke in the annealing grate, with the included crucibles or pots, is aglow. This operation has to be carefully watched, and requires judgment and experience so as to avoid over or under annealing, known among the men as "hot-nailing" or "slack nailing." The crucibles, in their red-hot condition, are then ready for use, that is, to be transferred to the furnaces or melting holes and the process described above.

The crucibles are generally only used twice, very occasionally three times, but in some works they are made to produce two rounds of high-speed

steel and one of carbon steel, or one round of high-speed steel, one of special carbon tool steel, and a third round of file steel, as may be required, the output of the works making high-speed steel being extremely varied.

The weight of an annealed pot as used in the high-speed steel processes detailed above is, on the average, about 30 pounds, occasionally more or less; whilst the quantity of material which these crucibles or pots can contain varies from 40 pounds to 70 pounds.—"Engineering."

COBALT-CROM, THE NEW STEEL ALLOY.

Widespread interest has been aroused by the announcement on behalf of Messrs. Darwin and Milner, of Sheffield, of the discovery of a tool steel stated to be equal in durability and hardness to high-speed steel, yet of which tungsten is not a component. "Cobalt-crom," as the new steel is called, is the outcome of much experiment and labor, extending over many years, on the part of the head of the firm of Darwin and Milner. He had long been experimenting in the direction of finding a steel which could be made without tungsten but would possess the necessary hardness of the so-called "high-speed" steels. About twelve months ago he discovered that by adding cobalt to chromium-carbon steel he converted a steel which had no appreciable red-cutting hardness into one which had this very valuable qualification.

It is a fairly well known fact that while tungsten high-speed steel has for some years past been extensively used for turning, planing and slotting tools, it has not been used so largely for milling cutters, taps, reamers, etc. It is estimated by Messrs. Darwin and Milner that of all the milling cutters, etc., used in the engineering world, at least 90 per cent are still being manufactured from carbon steel, whereas, on the other hand, the reverse is the case with turning tools, nine-tenths of which are made from tungsten high-speed steel. In the opinion of the firm the reason that high-speed steel has not been applied largely for milling cutters, taps, etc., lies in the difficulty which the tool maker experiences in hardening such tools having fine edges at the high heat required to secure from tungsten high-speed steel its best cutting efficiency. Tungsten high-speed steel requires hardening at from 1,250 degrees to 1,350 degrees C. if the best results are desired. To try and obtain this heat for milling cutters is very risky, and to harden at a lower heat gives less satisfactory results.

When the war broke out Sheffield firms received instructions that they must make milling cutters and taps from the very best high-speed steel, but there was endless trouble through the hardening process alone, with the result that carbon steel was soon substituted. Now it has been found that the maximum heat necessary for the hardening of the new steel is only 1,000 degrees C. As a matter of fact, if this heat is exceeded the tools give adverse results. This temperature allows of the manufacture of fine-edged tools of a perfectly satisfactory character with the red-cutting hardness of the tungsten high-speed steel. Hardening in practically all applications is stated to be satisfactory when the tool is allowed to cool naturally in still air free from drafts and currents, whereas with the tungsten high-speed steel, hardening from the extreme high heat renders it necessary to quench either with a strong air blast or a sudden immersion in liquid. Messrs. Darwin and Milner state that with the new steel it is possible to get absolutely the same standard of hardness throughout.

Samples of the steel have been tested in a large number of important works throughout the country, and have shown that the cobalt-crom is specially suitable for milling cutters, twist drills, reamers, taps, automatic forming tools, screw cutting, finishing tools generally, for machining gun-

metal and yellow metals, as well as for high-endurance drawing and blanking dies, lathe centers, shear blades, aeroplane and automobile combustion-engine valves, and every description of gages and instruments which have to resist abrasion. A test has been made showing that with a carbon steel blade saws can cut for half a day, with a tungsten high-speed steel blade for two days, but with the new steel four days without the need of grinding. We are informed also that particulars have come to hand of tests made in America between the new steel and the Becker iridium-cobalt-tungsten high-speed steel, these proving the tungstenless steel to be the equal of the famous American product.

Messrs. Darwin and Milner have applied for patent rights the world over, including by permission of the Patent Office authorities, Germany and Austria. A point which should not be overlooked is that the cutting efficiency of the new steel is quite equal, even in the form of castings, to that of tools made from the forged or rolled bar in which high-speed steel is supplied commercially, and as the material in the molten state is much more liquid than high-speed steel, it lends itself to all forms of tool-form castings. We are informed that the makers are preparing to put into operation an organization to supply orders for milling cutter blanks in this new steel within 48 hours, and this should be of great assistance to engineering concerns in times like the present.—"Page's Engineering Weekly."

THE USE OF VANADIUM IN STEEL MAKING.*

By G. L. Norris.

This discussion is confined almost entirely to the consideration of pearlitic-vanadium-ternary steel, or simple carbon-vanadium steel, and will not more than briefly touch on the quaternary-vanadium steels. First, however, the writer wishes to touch briefly upon a still somewhat prevalent popular idea that vanadium is a powerful scavenger, and that the beneficial effects of its use in steel are principally due to its removal of minute, residual amounts of oxygen and nitrogen from the steel. It has even been advanced by some individuals that when all the vanadium added has been completely used up in scavenging, and none remains in the steel, all the improvement or beneficial effects possible have been accomplished; despite the fact that there is an increase in the mechanical properties of the steel with increasing amounts of vanadium present, and that vanadium has equally as great beneficial effects in steels made under reducing conditions such as crucible and electric-furnace steels, as in the case of steels made under oxidizing conditions like open-hearth and Bessemer steels. While it is true that vanadium oxidizes readily and will combine with nitrogen, its value as a scavenger is negligible, as there are much cheaper metals that are as effective, or perhaps more so.

The remarkable effects of vanadium on steel are due entirely to its presence in the steel as an alloying element, and its influence on the other constituents with which it is in combination. When added to steel it is found in both the main constituents, ferrite and pearlite, but principally in the latter. Only a few hundredths of one per cent of the vanadium combines with the ferrite. This minute amount, however, appears to increase the strength, toughness, hardness and resistance to abrasion of the ferrite. Nearly all the vanadium, however, is found in the pearlite, in chemical combination with the cementite, as a compound carbide of vanadium and iron in the case of ternary steel, and as more complex carbides in the case of quaternary steels.

* Read before the American Society for Testing Materials.

Vanadium replaces the iron in the cementite or the carbide by increasing amounts until finally, when the percentage of vanadium is about 5 per cent all the iron has been replaced by the vanadium. The vanadium-containing cementite is not as mobile as ordinary cementite and consequently does not segregate into as large masses, but occurs in relatively minute particles and, therefore, is more uniformly distributed. It does not, consequently, readily occur as lamellar or thin plates in the pearlite, but in a granular or sorbitic condition. This strong tendency of vanadium to form sorbitic and even troostitic pearlite, is doubtless one of the reasons for the mechanical superiority of steels containing vanadium, not only statically but dynamically. Vanadium carbide is not as readily soluble on heating as is iron carbide, and consequently vanadium steel requires a higher temperature to dissolve the cementite and put the steel in the austenitic condition for quenching.

The effect of vanadium on the physical or mechanical properties of steel increases with the percentage of vanadium until 1 per cent is present, after which there is a decrease, even in the case of quenched steels, and with 3 per cent or more of vanadium the steel is actually softened on quenching until unusually high temperatures are reached, say about 1,300 to 1,400 degrees C.

Vanadium-steel hardenite has a greater thermal stability, or power to withstand elevated temperatures without softening or breaking down or separation of cementite. This property is responsible for the great improvement in high-speed steels, through the addition of vanadium to stand up longer under the high temperatures developed at the point of the tool in making heavy cuts at high speed. Percentages of vanadium as high as 3.50 have been successfully used in high-speed steel, and 1.50 to 2.50 per cent are not uncommon, although only a few years ago the percentage ranged from 0.30 to 0.75, and it was thought that the addition of over 1 per cent gave very little additional advantage. The improvement in high-speed steel through the use of vanadium has borne an almost direct relation to the percentage of vanadium present, and is considered to be from 60 to 100 per cent.

In the case of carbon-vanadium tool steel the use of vanadium has proved almost equally beneficial, although at present only about 0.2 per cent of vanadium is used in such steel. It has a wider quenching range—that is, can be heated higher without injury—hardens deeper, retains cutting edge longer, and is very much tougher and stronger. A bar of 1 per cent carbon tool steel containing 0.25 per cent vanadium, quenched and drawn back at 400 degrees C., will bend 90 degrees without failure, whereas a similar steel without vanadium will bend only about 20 or possibly 30 degrees. Comparative compression tests of such tool steels with like tempering or drawback gave on 1¼-inch cubes, 490,000 pounds for the vanadium steel and 278,000 pounds for the steel without vanadium. For battering tools, such as pneumatic chisels, sets, caulking tools, rock drills, etc., vanadium tool steel possesses marked superiority on account of its combination of hardness, strength and toughness.

One of the principal applications of vanadium steel has been for steel castings, particularly for locomotive frames. The composition of the steel is the same as usual for such castings, excepting for 0.15 per cent or more vanadium. The addition of this small amount of vanadium increases the elastic limit of the annealed castings 25 to 30 per cent without lowering the ductility. The tensile strength is not increased proportionately in the case of thoroughly annealed castings, but is usually 10 to 15 per cent greater. The following averages of tests within the same ranges of composition, having about 0.25 per cent carbon, 0.62 per cent manganese, 0.27 per cent silicon, with about 0.18 per cent vanadium in the vanadium steel, are typical:

	Carbon Steel.	Vanadium Steel.
Elastic limit, pounds per square inch	36,495	48,210
Tensile strength, pounds per square inch	73,820	79,930
Elongation in 2 inches, per cent	27.3	26.6
Reduction of area, per cent	46.0	48.1

For reasons apparent from what has been previously stated, vanadium-steel castings require a somewhat higher annealing temperature than ordinary steel castings. They are also more susceptible to hardening, and therefore should be cooled slowly in the annealing furnace. The annealing temperature should be about 875 degrees C.

There is a growing tendency to heat-treat and quench and temper steel castings for many purposes. Vanadium steel is much more suitable for heat-treatment than ordinary steel, as it hardens more on quenching and consequently much higher physical properties can be obtained. Even air cooling from the annealing temperature, followed by an annealing at a low temperature, greatly increased the elastic limit without affecting the ductility. This treatment meets the following minimum physical requirements with vanadium steel of the average chemical composition previously given:

Elastic limit, pounds per square inch	65,000
Tensile strength, pounds per square inch	80,000
Elongation in 2 inches, per cent	20
Reduction of area, per cent	40

Tests of quenched and tempered castings of the same composition show as follows, the quenching and drawback temperatures being the same for both steels:

	Carbon Steel.	Vanadium Steel.
Elastic limit, pounds per square inch	58,630	72,860
Tensile strength, pounds per square inch	90,630	95,630
Elongation in 2 inches, per cent	25	26
Reduction of area, per cent	52	56

For higher carbons the advantage in favor of vanadium steel is even greater. Water was the quenching medium.

There is also, doubtless, a great future for vanadium-quaternary-steel castings, both annealed and heat-treated, particularly the latter, for, as in the case of forged or rolled quaternary steels, the improvement in the mechanical properties from the presence of vanadium would be much greater even than in the case of simple carbon-vanadium steel.

The value of vanadium in simple carbon forging steels has been overshadowed by the greater mechanical properties of the vanadium-quaternary steels, such as chrome-vanadium, yet they have mechanical properties equal to those of ordinary 3 per cent nickel steel under like conditions excepting where the very high physical properties obtainable from quaternary steels are desired, carbon-vanadium steel can be used to advantage, especially for large forgings. This steel prevents fewer manufacturing difficulties than quaternary steels. It is less liable to losses from shrinkage cracks and checks in the ingot, and to heating and cooling strains in the forging and heat-treatment operations. It requires no more special care in handling than ordinary carbon steel, and is worked with equal facility.

Carbon-vanadium forging steel in the normalized condition, which may be described as annealed for grain refinement, has physical properties superior to those specified for heat-treated carbon-steel forgings. This simple treatment alone, therefore, gives physical properties sufficiently high for a great number of forgings that would otherwise have to be quenched

and tempered. A large field for annealed or normalized carbon-vanadium steel is its use for locomotive forgings. It is generally conceded that there is need of a steel of greater strength, not only to meet present conditions, but also to permit of reducing sections of reciprocating parts to obtain better counter-balancing. To meet this requirement the railroads several years ago turned to heat-treated carbon and heat-treated alloy steels, notably chrome-vanadium.

The use of heat-treated locomotive forgings has not proved altogether satisfactory for a number of reasons. One objection to the use of these forgings is the lack of heat-treating equipment in most railroad shops. This operates particularly in repair work where for any reason the forging has to be locally heated to straighten, stretch or shorten, destroying thereby the effect of this original heat treatment and producing inequalities which may result in failure. Probably no other class of forgings is subject to as abusive use as locomotive forgings, and under such conditions ordinary heat-treated forgings have not been found reliable. For these and other reasons annealed forgings are preferred. Consequently, a steel that could give in an annealed condition physical properties equal to or even better than those specified for heat-treated carbon-steel forgings, would prove very desirable. Carbon-vanadium forging steel responds to heat treatment in a very superior manner, and in this condition will meet the requirements specified by the society for quenched-and-tempered nickel-steel forgings.

As previously stated, the influence of vanadium on the mechanical properties of quaternary steels is even greater than on simple or carbon steel. The best known and most extensively used of these is chrome-vanadium steel. Nickel vanadium steel, while having possibly even higher tensile properties than chrome-vanadium steel, is considerably more expensive and has not been found to meet all conditions as satisfactorily as chrome-vanadium steel. It does not appear to have as high a resistance to shocks and repeated stresses. Nickel-chrome-vanadium steel has found extensive application in light armor such as deck plate, gun shields and armor-piercing projectiles, where its use has greatly improved the efficiency of these materials.—"Page's Engineering Weekly."

POSSIBILITIES OF DOUBLE HEAT-TREATMENT OF STEEL.

By Prof. Bradley Stoughton, Secretary of American Institute of Mining Engineers.

The time has arrived in America when we are obliged to lay more emphasis than ever on the question of quality and higher standards for steel for special work. We have nearly reached our limit of quality in cheap steel, as far as chemical and mechanical manufacture is concerned, but in the field of heat-treatment there is a great deal that can be done and without much expense. Most steel workers have had tools burst when quenching and are familiar with the hollow that frequently forms down the entire center of a tool due to the stresses in cooling. Casehardened articles are particularly liable to these stresses. It is the same way with armor plate, which is casehardened to perhaps as high as 2.5 per cent carbon on the surface but is down as low as 0.25 per cent in the back; this naturally causes all sorts of stresses, particularly in curved plates. One famous English metallurgist heats his armor plate to about 1,000 degrees C., cools it very slowly to about 700 degrees, and quenches it; then in order to get it fibrous he reheats it to about 650 degrees C. and quenches it.

That heat-treatment changes the molecular structure of steel is shown by the fact that any steel heated to about 775 degrees C. will lose its mag-

netism, and magnetism is usually admitted to be a molecular change. Yet the magnet steel invented by Sir Robert Hadfield, which contains about 2.75 per cent silicon and has as little of any other ingredient than iron as possible, when subjected to a double heat-treatment is changed to the most magnetic substance we know; it is even more magnetic than chemically pure iron. One of the most important results of heat-treatment is to change the size of grain for the reason that, every other condition of the steel being the same, if the grain size is small, the steel will be strong and tough, while if the grain size of the same steel is large, the steel will be weak. The object of almost all double heat-treatments is to produce a small size of grain in steel, for the highest strength and ductility is invariably accompanied by the smallest grain size.

The large size of the ferrite grains produced in one of the heat zones has given rise to many double heat-treatments, but it has not been understood as a rule. The only way to do the thing is to heat to between 700 and 900 degrees C., depending on the carbon content of the steel, and quench as rapidly as possible; then heat to just below 700 degrees C. and quench again. This is a common double heat-treatment for all steels of medium carbon that are to be of high quality, because in these medium-carbon steels the ferrite forms a large portion of the mass, and if that can be obtained in small grains, a good quality of steel is cheaply obtained from a steel that would otherwise be much inferior. Steel containing approximately 0.30 to 0.35 per cent carbon and 0.10 to 0.15 per cent vanadium, without chromium or nickel is commonly heated to 900 degrees C., quenched, heated to, say, 600 degrees and quenched. If it is desired to have the steel a little stronger and not quite so ductile, the second heat should come to 500, 400, or even 350 degrees, depending on the amount of strength desired. Vanadium also has the effect of making the absorption of the ferrite in steel take place very slowly; consequently it is necessary not only to heat slowly to 900 degrees, but to hold the metal there for some minutes in order that all the ferrite will be absorbed. Some advise heating above 950 degrees, cooling to 875 degrees, and quenching. Exactly the same result is obtained by heating a little more slowly to exactly 900 degrees, or else heating to 950 degrees and maintaining that heat a while.

Axles are heated to about 720 or 730 degrees, quenched, heated just below 700 degrees, and quenched again. Axles containing chromium and vanadium are heated a good deal higher, quenched in oil, and heated above 700 degrees, which heat is maintained for about twenty minutes and then allowed to cool slowly. The chromium has so strong a tendency to retain the carbon dissolved in the steel that if the steel is quenched it will be too brittle.

Casehardened gears are cooled to atmospheric temperature, then heated to between 700 and 900 degrees C., depending on the carbon content, quenched, heated above 700 degrees, and quenched. As the core of the gear contains only a small amount of carbon, so that it is rich in ferrite, the first quenching toughens the core by bringing it rapidly through the ferrite range, thus preventing the ferrite grains from increasing in size. The first quenching gives a tough core and a brittle case, and the second quenching gives the fine-grain case. As this is quite brittle, it is usually annealed by heating to about 200 degrees C.

One reason the double heat-treatment has been neglected is that when steel is heated up to the austenite range an enormous size of grain is produced. Steel heated up to this point has been known as burnt and as overheated steel, but it is possible to remove any bad effects of that treatment. The steel is heated to this high temperature and quenched, and then all the bad crystallization is removed by heating, say, to 1,100 degrees C., quenching, heating to 800 degrees and quenching, and finally heating to 700

degrees and quenching. If ordinary commercial, Bessemer, low-carbon steel is heated to 1,400 degrees C., quenched in water, heated to 750 degrees, and quenched in water, its strength will be increased from two to four times without decreasing the ductility more than 25 per cent. If a little more elaborate process is used, a steel over 200,000 pounds per square inch tensile strength with a good degree of ductility will be obtained at only a slightly greater cost than ordinary Bessemer bars or rods.

Vanadium keeps the cementite from segregating into coarse grains, and to some extent keeps the austenite from coarsening in this range. It is for that reason that vanadium has been used in high-speed steel, for these steels may then be heated to a very high temperature without acquiring such a coarse grain. The result has been an increase of 60 to 100 per cent in the service of the steel. Nickel has a similar effect, and 12.5 per cent nickel-steel will stand considerable overheating in the austenite range without coming to as much harm as straight carbon steel. When the steel contains 25 per cent nickel it can be heated to almost any point without getting coarse crystallization. Vanadium exerts its influence chiefly on the cementite, and nickel chiefly on the ferrite. Most of the nickel dissolves in the ferrite in steel, whereas most of the vanadium dissolves in the cementite, so that a small amount of nickel is exceedingly advantageous in steels to be heat-treated in the ferrite range where the ferrite is to be kept small. Manganese has the same effect as vanadium and nickel in keeping the grains of austenite small. Nickel-vanadium steel, manganese-vanadium steel, chromium-vanadium steel give many combinations that afford an almost endless line of possibilities.

By cooling steel to liquid-air temperatures, Sir Robert Hadfield has made bars of manganese steel and 25 per cent nickel steel that are strongly magnetic on one end and as non-magnetic as lead on the other. All these steels are normally non-magnetic, but by cooling to the temperature of liquid air that end becomes magnetic. This opens a line of possibilities with heat-treatment that as yet have scarcely been touched.—"Machinery."

MANUFACTURE OF ELECTRIC TOOL STEEL.

By E. A. Suverkrop.

(Abridged.)

The large electric steel plant of the Ludlum Steel Co. started the manufacture of crucible tool steel in a small plant at Pompton, N. J., in the year of 1854, and in 1907 moved to Watervliet, N. Y. Until 1910 the concern devoted its entire attention to the production of crucible steels. There is a story that the writer was told by a disinterested party, which seems to be a fitting beginning to this. Some twenty years ago a steel maker sent one of his skilled men to get a job at the old Pompton plant with the instruction to "find out what they put in their steel and how they make it so much better than we do!" The story goes that after a few weeks he sent in a report to this effect: "Their mix is exactly the same as ours, but they turn out better steel because of the care exercised at all stages of manufacture." This critical care in the selection of material, in the manufacture of the product and in the elimination of defective material is impressed on the investigator at every step.

High-grade tool steel is produced by two processes: (1) the crucible, (2) the electric furnace. The entire output of the Ludlum Steel Co. is now the product of electric furnaces (which not only produce a better steel but produce it at a lower cost), in order that if possible even greater care can be devoted to the elimination of faulty material, without adding materially to the ultimate cost of the finished steel.

Crucible steel is made by remelting very high-grade, pure scrap. This is accomplished by the use of a plumbago or clay crucible which is filled with short pieces of Swedish, Walloon (Belgian), or Styrian charcoal iron. A lower grade of crucible steel is made from puddled iron, and a great many crucible steel manufacturers, in this country, because of the difficulties of shipment at the present time, are forced to make their crucible steel from puddled iron. Puddled iron is made from low-phosphorus and low-sulphur pig iron, melted in the presence of an extremely oxidizing flame, which has the effect of oxidizing out most of the impurities. The remaining semi-plastic metal is then rolled into balls. These are full of slag, oxygen and nitrogen. They weigh about two or three hundred pounds each and are put through a mechanical squeezer which is a rotary mill with external and internal teeth. The internal drum rotates, and, due to its eccentric arrangement with the shell, gradually turns the ball around and squeezes it out into an irregular shaped mass. This material is then squeezed further by being rolled in a rolling mill. The slag by this time having been almost entirely squeezed out of the bars, the bars $3 \times \frac{5}{8}$ inches are cut into short lengths and used as the charge in the crucible to make crucible steel. The charge consists either of charcoal-refined iron or puddled iron—the additions of carbon, manganese, alloys, etc., being arranged with the charcoal iron in the crucible. The crucible is placed in a coke, oil or producer-fired furnace and heated until the charge melts. The melter, who must be a very highly skilled man, periodically examines the melt, observes its progress, and waits until it is quiet in the crucible; that is to say "killed," the melt then being known as "dead."

The metal is poured from the crucible into a ladle, or else direct from the crucible into ingot molds. Because the crucible process is purely one of remelting, a refining of the steel cannot take place. In other words, the quality of the steel is dependent on the quality of the ingredients.

ELECTRIC STEEL MANUFACTURE.

The electric crucible-steel furnace is capable of completely refining the steel; that is to say of removing from it the undesirable metalloids such as phosphorus, sulphur, excess of silicon, manganese and carbon; also the gases: oxygen, hydrogen, nitrogen, etc.

The following report of the working of the electric furnace for the production of twist-drill steel will give the layman an idea of how closely the melt is watched. It is doubtful if any such detailed report has ever been printed.

Heat started 4:55 A. M.—Part of the charge was loaded into the furnace and melted; the remainder was added after the metal had been made molten since the charge, being of a bulky nature, could not be entirely charged into the furnace at the start. A quantity of lime was added to the cold metal so as to keep the surface of the molten steel covered and protected from atmospheric influences; also to give the correct dephosphorizing slag. One hundred pounds of mill scale were added to the charge, to render the slag extremely oxidizing.

9:16 A. M.—The melt was observed to be boiling but the slag too thick. Two shovels of silica sand were added to thin the slag. The slag was well raked thoroughly to mix into the sand.

9:24 A. M.—A sample was taken for the laboratory. The metal was then fairly quiet, the slag was extremely oxidizing and black. It is necessary that the color of the slag be kept black, as this is the indication of an oxidizing slag produced by iron oxide.

9:30 A. M.—Lime was added for the purpose of thickening up the slag so as to make the operation of raking off this slag comparatively easy.

9:35 A. M.—The current was turned off and the black dephosphorizing

slag was raked off. The analysis of melt showed carbon 0.59 per cent, chromium 0.37 per cent, phosphorus 0.018 per cent, sulphur 0.025 per cent.

9:39 A. M.—Ten shovels of lime and fluor spar and sand were then added to the melt to form a second slag. The current consumption was 7,500 ampères per electrode.

9:45 A. M.—Sample of the slag was taken showing the slag thin—which is the ideal consistency. The lime was not entirely fused.

9:53 A. M.—A further sample of the slag was taken, showing the same to be extremely desulphurizing and reducing. The color of the slag was gray, condition smooth and thin.

9:54 A. M.—A sample was taken of the metal for laboratory analysis. The current was then turned off.

10 A. M.—The temperature of the slag with the current off was 2,573 degrees F.

10:02 A. M.—The current was turned on.

10:03 A. M.—Five shovels of the mixture in the boxes were then added to the slag to augment its desulphurizing nature.

10:08 A. M.—A further sample of the slag was taken, showing the color gray, and of good desulphurizing and deoxidizing condition.

10:10 A. M.—Lime was added to thicken the slag so as readily to remove same. Current turned off at 10:12 A. M.

10:14 A. M.—Slag was very thick and easily raked off. Temperature of the metal was then taken; 2,570 degrees F.

10:20 A. M.—A predetermined amount of carbon was thrown into the melt to add to the carbon and further degasify the steel.

10:22 A. M.—The doors of the furnace were luted up with clay so as to prevent the atmosphere attacking the surface of the molten steel.

10:40 A. M.—The furnace was unsealed, the temperature of the steel was 2,596 degrees. A third slag was immediately added by throwing in seven shovelfulls of the mixture of lime, silica sand and fluorspar.

10:52 A. M.—A sample of the slag was then taken which showed the color gray and in excellent condition. Correct amount of alloys was then added to bring the steel up to the required analysis.

10:59 A. M.—A sample of the slag was taken which had a slight greenish color due to the chromium.

11:02 A. M.—Slag of metal was sent to the laboratory for analysis. The temperature of the slag was 2,662 degrees.

11:05 A. M.—The current was turned off. Sample was taken of the slag. The green color had almost disappeared, showing the chromium present in the slag had returned to the metal.

11:08 A. M.—Temperature of the slag was 2,652 degrees. It is noted that the temperature has now fallen slightly. Current turned off waiting analysis. The ladles were now rapidly heated in preparation of the melt being teemed.

11:32 A. M.—While still waiting for the analysis, four shovels of lime were added to the slag, as it was noticed the slag was getting a little thin.

11:35 A. M.—A sample was taken of the slag; outside brown and inside gray, excellent condition.

11:42 A. M.—Laboratory analysis returned. A small quantity of alloys added to finally adjust the chemical analysis of the steel.

11:52 A. M.—Test bar of the metal forged out, to find the mechanical working qualities.

11:53 A. M.—The metal agitated so as to insure complete mixing of the alloys.

11:55 A. M.—The melt held to insure saturation of the added alloys, sample again taken of the slag, small quantity of lime added, to keep the slag in a neutral condition, it now being neither oxidizing nor reducing. Temperature of the slag and the metal were then taken, 2,650 degrees F.

11:58 A. M.—Small quantity of ferro-silicon was added to the melt to remove gases. It was noted that the metal did not boil at all, the surface of the metal being entirely still.

12:06 P. M.—A small quantity of fluorspar and silica sand were added to thin the slag and keep it neutral. Temperature of metal and slag was then taken, 2,690 degrees F. Current was then turned off and the metal was ready to be teemed into the ladles. The metal was poured from the furnace into the ladle; temperature was then taken, 2,641 degrees F. The ladle was put onto its car and placed over the first ingot; temperature at the first pouring of the ingot, 2,600 degrees F. Time required to pour first ingot, 35 seconds. Seventeen ingots were cast from first ladle; temperature of the last ingot, 2,550 degrees F. Second ladle was then used and the remainder of the melt in the furnace teemed into the ladle. Temperature of the metal pouring from the furnace into the ladle, 2,618 degrees F. The ladle was then placed in position over the ingot molds; first ingot took 31 seconds to pour; temperature of the metal 2,600 degrees F. Thirty-five ingots were cast from this ladle; the last ingot poured, the temperature of the metal was 2,575 degrees F.

The correct analysis of this steel when in ingot form was carbon 0.95 per cent, chromium 0.45 per cent, silicon 0.15 per cent, manganese 0.20 per cent, phosphorus 0.015 per cent, and sulphur 0.012 per cent.

SELECTION OF MATERIALS.

A careful analysis is made of all raw materials received at the plant. Raw materials in which the sulphur and phosphorus contents are below 0.03 per cent is used in the manufacture of tool steel. That in which the percentage of sulphur and phosphorus is above this figure is used in the manufacture of low-phosphorus pig iron, to which a department in the works is devoted. Owing to the possibility of refining the metal in the electric furnace, a raw-material charge of known analysis is melted down, with a slag covering the surface of the molten steel at all times. The first slag is of lime with a small amount of sand to keep it thin, and is known as an oxidizing slag since it oxidizes out the impurities: phosphorus, some sulphur, silicon, manganese and carbon. The phosphorus comes up into the slag as phosphoric acid and changes to calcium phosphate. Great care must be exercised during this operation; the temperature must be definitely controlled; if too low the phosphorus is not taken out as speedily as desired; on the other hand, if the temperature is too high the phosphorus is reabsorbed in the melt, and if once reabsorbed is very difficult to eliminate. While the condition of low temperature which is favorable to the elimination of phosphorus prolongs the melting operation, it does not in any way injure the melt.

The period during which dephosphorization is most active is that preceding the actual melting of the charge.

The electrodes in the headpiece of the furnace are gradually lowered, melting their way through the charge. The metal as it becomes fluid, percolates to the bottom of the furnace where it forms a pool. The heat of the arc, direct, and that reflected by the roof of the furnace, gradually melts the charge which augments the pool in the bottom of the furnace. When the charge is melted the first slag is raked off the steel, examined for color and texture, as these convey to the experienced furnace-man positive information regarding the melt. A second lime slag is then added; this carries with it a definite quantity of fluorspar and silicious material, generally sand. This second slag should be very "thin" as known in the terms of the melting shop. Fluorspar effectively helps to remove the sulphur, oxygen, hydrogen and nitrogen gases, and thins the slag. The silicon also thins the slag. If lime alone were used, the slag would be too thick and

the chemical reactions taking place would be correspondingly slow. This second slag is known as a reducing slag, the oxygen rises to the surface and oxidizes the carbon or silicon. The sulphur rises and mixes with the slag and forms calcium sulphide, a comparatively stable compound. Great care must be exercised that the calcium sulphide is not thrown back into the steel by being oxidized and forming calcium sulphate. The action which takes place when the sulphur is eliminated from the metal is: iron sulphide + carbon + calcium oxide, which transforms into calcium sulphide. The iron passes back into the melt while the carbon passes out of the furnace in the form of carbon monoxide. The manganese sulphide + carbon + calcium oxide changes to calcium sulphide which enters the slag, while the liberated manganese returns to the melt and the carbon monoxide passes out of the furnace in the form of gas.

The slag is tested frequently during the process of refining, and the color of the slag is carefully noted. When the slag has done its work it becomes white, which, however, does not show that the action is completed, but merely that that particular slag has done its work. The slag having become white, the melter, without making any analysis, knows that it has reached the desired state. The sulphur in the slag and in the melt is checked, while the process of slagging is repeated and the melter is informed as to the chemical content of his melt. The test pieces are obtained by means of an iron bar which has a spoon-shaped end about 2½ inches in diameter.

During the process of refining, the temperatures of the molten steel in the furnace are very carefully taken. The melt after having been passed by the laboratory is ready to have the second slag removed by raking it off in a manner similar to the first. A third slag of lime and fluorspar is added, but without silicon or carbon. This slag acts as a blanket to keep the steel from oxidizing. The addition of alloys which are to bring the steel up to the desired chemical analysis is then made. Sufficient time is allowed for these alloys thoroughly to mix in the melt. The molten steel is then ready to be teemed; that is, poured out of the furnace into the ladle. When the ladle is about one-quarter full, some vanadium is thrown in further to deoxidize the metal. The third slag is not removed from the furnace, but is allowed to pour out with the steel into the ladle.

It is necessary that in pouring the molten steel from the ladle into the ingot molds the slag shall not be carried down by the metal, therefore the ladle is bottom-poured. There is a hole in the bottom of the ladle which is called the nozzle, and the necessary refractory plug is fitted into the top of this hole or nozzle, this plug being called the stopper. The ladle nozzle is then placed over the center of each ingot mold in turn, and the metal allowed to run into it. The temperature at which the metal is poured is of prime importance and is carefully noted; also the temperature is noted when the last ingot is poured. By this means it is possible to determine the composition of the metal when it freezes—that is to say, becomes solid.

If the metal is poured at too high a temperature and a long time elapses between the period when the metal is poured into the ingot molds and the time when it freezes or becomes solid, the steel crystals which form will have a tendency to throw out their various alloys (all alloys with iron are impurities of iron, although some of these impurities are very desirable). Slow cooling sets up segregation, which is to say that the alloys have a tendency to segregate in the ingot more in one place than in another. This is particularly true of the low melting-point metaloids, under which heading come phosphorus and sulphur. Providing this segregation can be controlled, it forms another means of eliminating these undesirable alloys from the steel because they segregate at the top of the ingot, which part is subsequently cut off.

HAMMER SHOP.

High-speed steel is stripped from the ingot molds while still hot and run to the preheating furnace in the hammer shop. The temperature of the preheat is 1,600 degrees F., and the ingots remain here for a period of 5 to 6 hours. From here the ingots go to the high-heat furnace where they are heated to 1,950 degrees F., thence direct to the hammers where they are forged to size, care being taken that the temperature of the work does not fall below 1,750 degrees F.

The structure of the steel in the ingots as cast is coarsely crystalline. Any steel in this state is quite unfit for tools and must therefore be mechanically worked to reduce the grain (that is to say the crystal sizes) to extreme smallness. The smaller the crystals or grains the higher the refinement and better the steel.

Steel can be refined by heat treatment, but when heat treatment and mechanical work are carried on together the refining is much more active and more thorough. Therefore all ingots, before they can be used as tool steel, must be forged, and for convenience rolled into various shapes. The steel is heated very slowly and thoroughly soaked in specially designed furnaces, then carefully forged down (known as "cogged") under the hammer to certain definite sizes which can be conveniently handled in the rolling mill. This forging must be done very carefully, otherwise the steel may be ruined. The pressure applied to the surface of the steel under the hammer must be applied at right angles to the face of the hammer. The contact under the head of the hammer and on the anvil must be on a line parallel to the axis of the hammer, otherwise a shearing is set up in the ingot which, if excessive, results in a burst center. Too light hammering will also injure the steel, opening it up in the center.

FORGING FROM INGOTS TO BILLETS.

In forging the ingots down to billets, which is the state after the first mechanical reduction, it is essential that the ingot should be worked in square form to very nearly the finished size, and then if the bar is to be still further reduced to hexagon, octagon or round, the corners are reduced until the desired shape is obtained. If an ingot which is to be turned into a round billet or a round forging were forged at the start into a round shape the bar would have a split center.

The billets after being forged in the hammer shop, pass to the inspectors. The inspectors examine them for surface defects, seams, blemishes, cracks, etc., which are outlined on the billet with a yellow wax crayon which shows quite clearly on the dark ingot. The high-speed billets (which of course are too hard to chip) go to the grinding shop, where the defects are ground out.

Billets other than high-speed are inspected and marked in the same way, but as the steel is soft enough to chip, the defects are removed with the pneumatic hammer. The chisels used for this work are rounded so that round-bottomed grooves are left where the defects are chipped out. The grooves must be round-bottomed so they will roll out in the subsequent rolling operation.

From the inspection and grinding the steel is either returned to the hammers or taken to the rolling mill, very carefully reheated in specially designed furnaces and passed through cogging rolls (that is, the billet mill), after which they are still further reduced in the bar mill. After each reduction the work is inspected and, if necessary, chipped or ground as described. If this were not done the imperfections would be rolled still further into the bar, thereby ruining the product.

The rolling mill is styled as so many "stands," each of which is composed of two uprights with the rollers horizontally arranged. The mill is

also styled as "two high" or "three high," which is to say that there are two rolls or three rolls in each stand. In a two-high stand rolling mill the bar is passed through shaped rollers along the same plane; in a three-high stand the bar is rolled alternately between the bottom roll and the middle roll and the top roll and the middle roll.

The billets are cut in the shear to certain dimensions and are then inspected. The measuring stops on a bar-shearing machine are, for the sake of economy, so set that the length of the bar will be a multiple of a certain length designated by the purchaser.

The bar mill, also known as the merchant-bar mill because it is capable of rolling to the ordinary merchant-bar sizes, is used for reducing billets down to the finished size. The billets are carefully reheated (particular attention being given to the time which these billets remain in the furnace) to the temperature prior to rolling, the same care, of course, having previously been taken in the billet mill. The steel is then rolled in the bar mill, a certain definite reduction taking place at each pass. The reduction of cross-sectional area of the work amounts to from 15 to 20 per cent per pass. The large rolls of the billet mill are 18 inches in diameter, and have a surface speed of about 500 feet per minute. The small rolls of the bar mill are 10 inches in diameter, and have a surface speed of from 650 to 700 feet per minute. The roughing rolls are made of alloy steel, heat treated, and the finishing rolls of chilled cast iron. The finishing temperature, that is to say the temperature at which the metal leaves the last pass in the rolling mill, is very carefully controlled. This is important, as the grain size and the future effectiveness of the steel are very largely affected by the temperature at the last pass. The mill for rolling the smaller sizes of bar is called a wire mill, and work as small as 3/16-inch diameter is rolled in it. The wire is coiled while hot, as it comes from the last pass.

Each analysis of tool steel must be mechanically worked at a certain temperature, therefore the furnaces are heated for each class of steel to a certain temperature and this temperature is controlled by pyrometers. The use of pyrometers is not at all common in the steel industry, and in many mills the temperature is gaged only by the eye and judgment of the furnace man.

The furnace man is instructed with each and every batch of steel to heat the furnace to a certain temperature which is recorded by a pyrometer over the furnace door. The finishing temperatures are also controlled by an inspector by the means of an optical pyrometer. Every bar that is not up to the standard temperature is marked so that it may be especially examined and passed upon. In most instances, these special bars are scrapped and remelted.

FINAL INSPECTION.

After the steel has passed inspection in the rolling mills it is sent to the warehouse to be finally viewed, a fracture taken of every bar, and each bar inspected for mechanical faults, seams, laps, cracks and general imperfections. The smaller bars are nicked with a cold chisel and broken. Each end of each and every bar is nicked and broken, and if a defect shows in the fracture another piece is broken off. Each bar is inspected from end to end. Any suspicious looking mark on the bar is filed away to see whether it is a mere surface defect or a crack which penetrates the bar. A certain quantity of this material is then annealed, and the remainder, which is known as normal steel, is then stored in the warehouse ready for shipment to those customers who order the material unannealed.

After annealing, the steel bars, when required, are straightened in a mechanical straightener, and are carefully inspected, some being sawed and others fractured on the end so as to determine that the grain is of the de-

sired fineness. Of every 100 pounds of steel that goes into the furnace, only about 60 pounds passes the final inspection. All bars of steel that pass are then carefully stored in their respective bins so that no two qualities of steel can become mixed. The bars that are shipped direct from the warehouse are painted with various distinguishing symbols.—"American Machinist."

PRODUCING SYNTHETIC GASOLINE BY ELECTROCHEMICAL MEANS.*

By L. B. Cherry.

It is well known that most of the crude-oil distilates and natural gases of the United States are of the paraffin series of hydrocarbons and that natural gas is composed largely of methane, which occupies a place at the beginning of the series and is the simplest of the group; while going toward the more complicated molecular structures, we find about 60 well-defined, saturated, normal hydrocarbons, terminating with the solid hexacontane, with a melting point of 101 degrees Centigrade.

Our commercial gasoline occupies a place in the series a little above those which are normally gas, and is composed of a probable blend of hexane, heptane and octane, for a good grade of gasoline, although we are tempted to believe that some grades on the market today contain hydrocarbons of much higher boiling points.

The aforementioned hydrocarbons are called paraffins because they resist at low temperatures the action of most reagents, and are in fact compounds, with an almost indifferent character, refusing to take on more hydrogen or even to chemically unite with any of their associates.

If we could chemically unite, in proper proportion and form, sufficient methane, or hydrogen gas, to certain of the heavier hydrocarbons, we could produce synthetic gasoline. The small increase in percentage of hydrogen necessary to convert some of the heavier fractions of crude oil into the lighter and lower boiling-point compounds is surprisingly small. The first experiments along this line were conducted with a small laboratory equipment, the still being composed of a piece of 3-inch pipe, the ends of which were closed with caps, threaded and screwed on. The treatment chambers, wherein the gas and oil vapors were subjected to high-voltage oscillatory, high-frequency current, were made of 1-inch pipe which had been plugged and reamed, the ends of which were fitted with modified spark plugs which served to lead the current through a fine No. 30 platinum wire. This wire passed through the center of the pipe from end to end and, when connected to the high-frequency coil, subjected the gas and oil vapor passing through the pipe to a silent electric discharge alternating back and forth between the central electrode and the pipe wall, and at a right angle to the direction of the flow of gas.

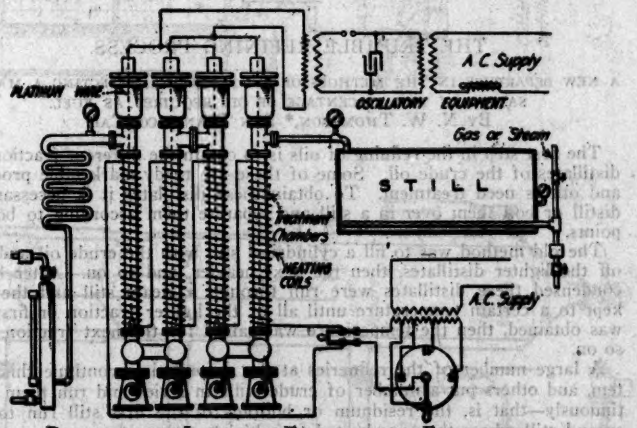
This first equipment was very crude, without pyrometers or pressure gages, and was operated at high pressures and high temperatures, but it served to demonstrate the feasibility of the process by converting 42.5-degrees B. kerosene into 65-degrees B. gravity gasoline. As already stated, this was done at high pressures and temperatures, and was operated for several hours before the experiment exploded, completely wrecking the small frame building in which it was located. Later experiments with larger and better equipment proved that a very high frequency was necessary to obtain low boiling-point compound and that increasing the frequency would raise the specific gravity (B.) and lower the boiling point.

*Abstract from a paper presented at the Thirty-second General Meeting of the Electrochemical Society, held at Pittsburgh, Oct. 3 to 6, 1917.

The still of this equipment was of 4-inch double-extra-heavy pipe construction, with welded flanges, electrically heated, and consumed about 40 kv.-a. at 60 volts, using 60-cycle alternating current. The oscillatory electrical generating apparatus for this plant consisted of 1 kv.-a., 110- to 25,000-volts step-up transformer with a variable reactance in the primary, and a variable condenser, rotary spark gap and oscillatory transformer. An equipment of this type is shown in the figure.

The difficulties encountered with operation under high pressures were so great that it was decided to carry on experiments under atmospheric pressure, in which it was found that as good products could be obtained as when operating under higher pressure, but the percentage of conversion was not so great, and the gravity was not quite so high, 33 per cent of the kerosene being converted to an average density of 54 degrees B. (specific gravity 0.761), and by rerunning the residue, which was practically the same density as the original kerosene, it was discovered that an additional conversion of 33 per cent was obtained for every time fractionation was made, and the remaining residue run through the still.

At the conclusion of these experiments it was decided to build a larger plant and incorporate in its design the benefit of the experience with the older ones. The step-up transformer was designed for 110–220 to 50,000–100,000 volts, and rated at 30 kv.-a., but with a real capacity of 75 kv.-a.



Diagrammatic Layout of Fractionation Equipment.

In this plant the high-frequency equipment is capable of producing high-frequency current from 15,000 to 250,000 volts and has a capacity of converting into gasoline about 60,000 gallons of oil in one day of 24 hours.

Experiments with the first plant, with its 3-foot treatment chambers, proved that they were not sufficiently long, so when building the second plant the treatment chambers were built with a total length of 10 feet. For experimental purposes a 7-barrel still is used, which is fired with natural gas, and the vapor outlet of the still is connected to the treatment chambers. These pipes are heated by induction, being wound their entire length, as shown in the figure, with 00 copper cable, electrically insulated with a special fireproof insulating compound. On operating this equipment a 78-per cent conversation was made from kerosene, and as this was not con-

sidered ideal, it was decided to increase the length of the treatment chambers to 34 feet, which should give an efficiency approaching 100 per cent conversion. At the time of preparing this paper this new treatment chamber equipment could not be operated on account of delay in the delivery of substation apparatus which is to furnish power.

The possibilities of this process are apparently very great, for by converting the distillate between the benzine fraction and that part in which the lubricating oils are found, it is possible to increase the output of gasoline 200 per cent.

One kilowatt of electrical energy used in the treating chamber will convert 1,000 gallons of distillate in 24 hours, or a 60-kw. plant should convert about 60,000 gallons per day. A total of 1,440 kw.-hr. is required for the oscillatory current to treat 60,000 gallons. It has been found by experiment that approximately 40 kw.-hr. of electrical energy are required for heating the treatment chambers while producing 60 gallons per hour, and if the ratio holds true for a 60,000-gallon plant, it would take 39,360 kw.-hr. for heating the treatment chambers while 60,000 gallons was being converted; this added to 1,440 kw.-hr. would make 40,800 kw.-hr. total energy required, which, at 0.6c. per kw.-hr., at which price electric energy can easily be produced at refineries, would make a total of $244.80, or about 0.4c. per gallon.—"Power."

THE TRUMBLE REFINING PROCESS.

A NEW DEPARTURE IN THE METHODS OF OIL DISTILLING, EFFECTING A MARKED SAVING IN THE PERCENTAGE OF OIL REQUIRED AS FUEL.

By N. W. Thompson,* San Francisco, Cal.

The first step in the refining of oils is to obtain the different fractions or distillates of the crude oil. Some of these are ready marketable products and others need treatment. To obtain these distillates it is necessary to distill or boil them over in a still and separate them according to boiling points.

The old method was to fill a cylindrical still with the crude oil and boil off the lighter distillates, then the next heavier, and so on. After being condensed these distillates were run through a steam still and the still kept to a certain temperature until all of the lighter fraction or first cut was obtained, then the temperature was raised for the next fraction, and so on.

A large number of the refineries at the present time continue this system, and others put a number of crude stills in series and run them continuously—that is, the residuum or bottoms of the first still run to the second still where they are heated to a higher temperature, and so on to the last still, from which the residuum flows to a cooler and then to the tank. This residuum is generally fuel oil.

In these stills there is a large volume of oil over the fires and also an expensive installation to build and maintain. There can be no seams in the bottom of the stills, necessitating very large plates, and it is necessary to have perforated steam pipes in the bottoms to agitate the oil so as to keep the bottoms from burning, etc.

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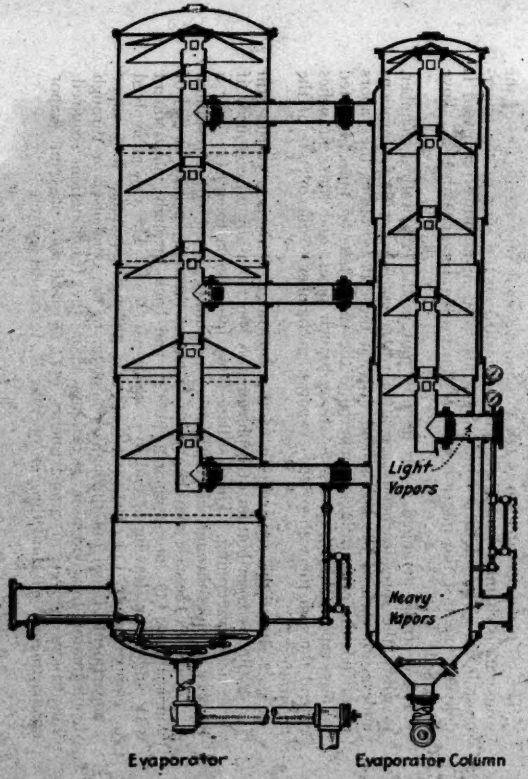

FIG. 1.—EVAPORATOR AND EVAPORATOR COLUMN.

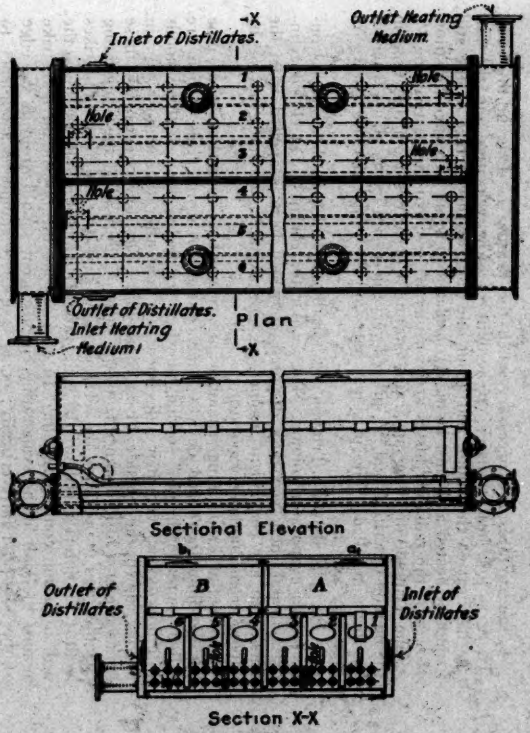

FIG. 2.—SEPARATOR.

NOTES.

Large surfaces are necessary when the heating is done in this way. There is low heat transfer on account of slow velocities and the low specific heat of the liquid, and about one-third of the surface of the still cannot be heated and must be highly insulated to avoid excessive radiation. However, the radiation losses are considerable even in the best settings.

The Trumble distilling apparatus is quite a departure from that used in the method described, and has proved very successful in the plants where it has been installed. The principal parts of the apparatus are shown in Figs. 1, 2 and 3.

In Fig. 1 is shown a Trumble evaporator, to which in this case is connected an evaporator column. The evaporator consists of a closed cylindrical metal shell, vertically disposed, to which heat is applied from the outside in any convenient manner, as, for example, by flue gases or by vapors from another evaporator.

Inside the evaporating chamber is arranged a central vertical pipe having umbrella-shaped devices attached thereto at intervals, which are called "spreader hoods," so that oil when fed to the apex of these hoods will flow down over their sides in a thin film after the analogy of rain flowing down the outside of an umbrella. The lower edges of these hoods are at a little distance from the sides of the wall of the evaporating chamber, and in operation the oil flows down and over the hood and strikes against the interior wall of the evaporating chamber and flows down the wall in a thin continuous film. In case any of the oil should not strike against the wall in the evaporating chamber, but should drop off the edge of the hood and fall vertically; or in case there should be a tendency to bubbling or foaming on the wall of the evaporating chamber, whereby a portion of the oil may be thrown back toward the center of the evaporating chamber, such oil will be caught by the next spreader hood and will flow down the surface thereof, thereby insuring an ultimate spreading of the oil on the wall of the evaporating chamber.

The oil to be operated on is fed through a supply pipe to the top of the evaporating chamber and discharges downward on the apex of the uppermost hood. The oil then flows down this hood in a thin stream and is delivered against the interior wall of the evaporating chamber as above described.

The centrally arranged vapor take-off pipe in the evaporating chamber, to which the spreader hoods are attached, is provided with perforations underneath each of the hoods, and through these perforations the vapors pass from the evaporating chamber into the vapor take-off pipe. Located in this vapor take-off pipe are lateral branch pipes, and these branches extend through the wall of the evaporating chamber to the outside of the apparatus.

Fig. 2 shows the construction of the separator. This is a re-run still for distillates from bottom of dephlegmators, or from any other source, if these need fractioning. The distillate flows into compartment (1) over a number of pipes through which a heating medium is passed, either residuum or vapors, as the case may be, and then passes through opening at end of compartment (1) into compartment (2) and back through compartment (3), and so on, leaving at end of compartment (6). The vapors evolved in compartments (1), (2) and (3) pass through openings into vapor compartment *A* and out through opening a_1 to the condenser or through dephlegmator to condenser, depending upon the fractionation required. The vapors evolved in compartments (4), (5) and (6) pass into vapor compartment *B* and out through opening b_1, to condenser, etc.

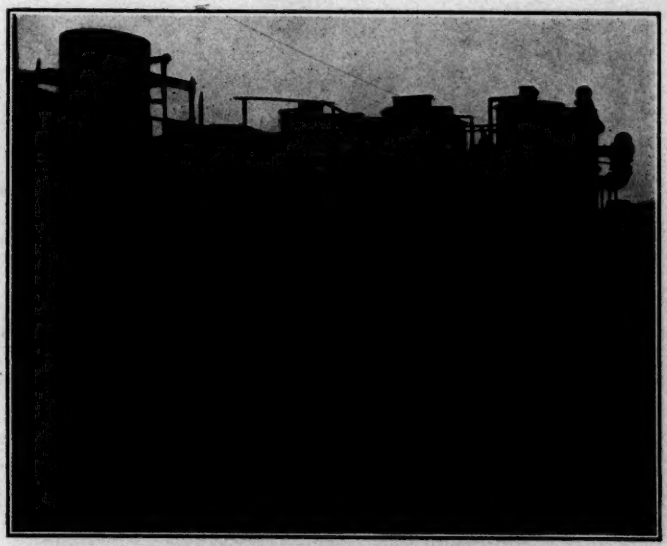

FIG. 3.—GENERAL ARRANGEMENT OF PLANT.

FIG. 4.—ARRANGEMENT OF RECEIVING BOXES.

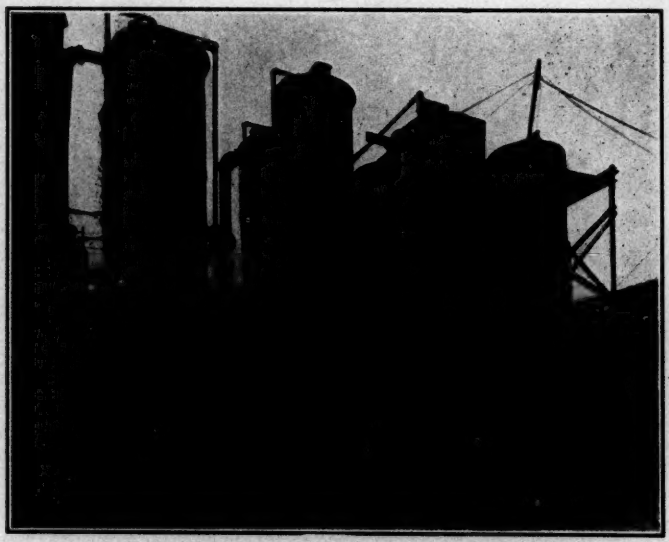

FIG. 5.—DEPHLEGMATORS WITH HEAT EXCHANGERS UNDERNEATH.

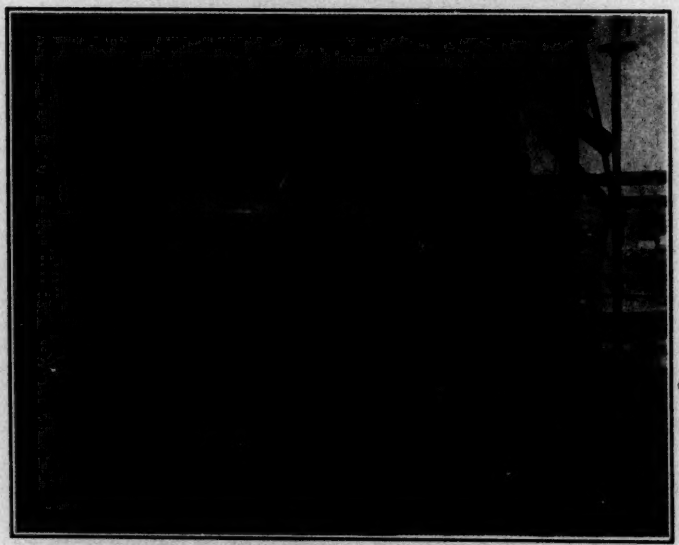

FIG. 6.—OPERATING VALVES FOR DISTILLATES FROM DEPHLEGMATORS.

The manifolds on the ends of the separators are provided with covers having stuffing-boxes through which valves are operated to regulate the flow through tubes in each compartment, thereby controlling the heat in these compartments.

The illustrations, Figs. 3 to 6, inclusive, are from photographs taken during the construction of a plant employing the Trumble system. Fig. 3 gives a general view. In the foreground, at the left, is the pipe heater, and located in the brick stack at the left is the evaporator. In the distance are the dephlegmators, in the foreground the separators and the vertical condensers for the vapors from the separators. Under the arches are the coolers for distillates. The pipe lines in the trench are the distillate lines from the condensers and coolers carrying the cooled products to the receiving boxes. The arrangement of these receiving boxes is shown in Fig. 4. Behind the boxes are dewatering traps for separating the water from the distillates. Fig. 5 shows the dephlegmators with the heat exchangers underneath. Fig. 6 shows the operating valves for the distillates from the dephlegmators. These valves as well as all operating valves are handled from one platform.

All plants are designed to meet the conditions under which they are to be used, but they are very flexible and are able to take care of a very wide range of conditions by controlling the heat of the furnace and the velocity of the oil which runs through it. It will be impossible to go into the many different arrangements of plant design, but the following will make clear the running of a typical and successful plant.

The flow of the crude oil through the plant is indicated by the flow sheet in Fig. 7. The oil enters through a 6-inch line and is used as a cooling medium in the six coolers shown. These coolers are of the horizontal tubular type, 30 inches in diameter, with 62 2-inch by 18-foot tubes. The oil enters at the bottom and passes through the tubes, making four passes, and comes out at the top and goes into a header, through which it passes into the four heat exchangers or coolers for the residuum.

These heat exchangers are 48 inches in diameter, and have 178 2-inch by 18-foot tubes. The crude oil enters the first heat exchanger at the bottom and passes through the tubes, making six passes, and out at the top into the bottom of the next exchanger, and so on, to the heater pipes, where it is split in two, each half passing in series through 72 18¾-foot lengths of 4-inch pipe, flowing back and forth and upward at all times, and then into the top of the evaporator, where it flows down the sides in a thin film.

The oil in passing through the heater pipes and evaporator is heated by the flue gases. The vapors evolved are separated in the evaporator and taken care of later. The oil is maintained at a constant level in the bottom of the evaporator and runs out of the bottom of it as residuum. A perforated steam coil is placed in the bottom of the evaporator under the liquid, and superheated steam is passed through it in order to drive off any of the lighter distillates which may drop back from the vapors. However, very little steam is necessary in this case, as the heat losses are supplied by the flue gases.

After leaving the evaporator the residuum is used as a heating medium for redistilling the distillates passing through the separators, and then flows through the heat exchangers counter-current to the crude oil. The residuum enters the first heat exchanger at the top and makes two passes around the tubes and out at the bottom, then into the top of the second exchanger, and so on through the five heat exchangers and through a standpipe, which is vented and which controls the head of residuum in the bottom of the evaporator, and then to the storage tanks.

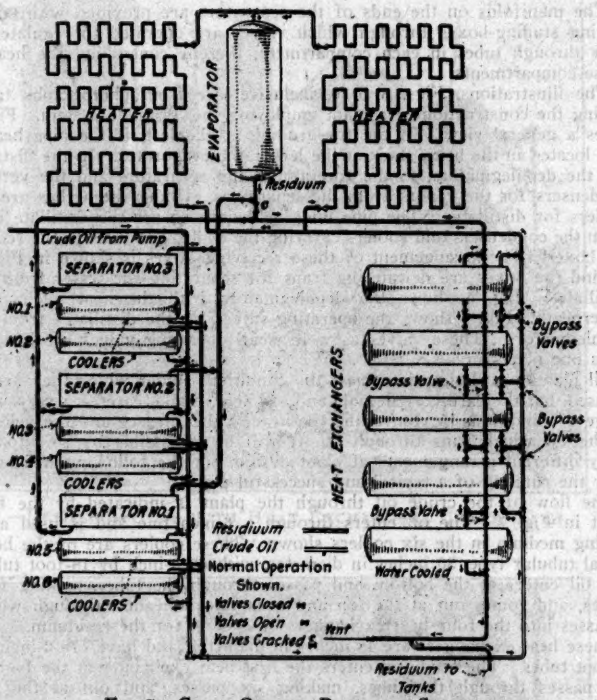

FIG. 7.—CRUDE OIL AND RESIDUUM SYSTEM.

Having followed the course of the crude oil through the apparatus, showing the separation of it into vapors and residuum, it remains to dispose of the vapors shown on flow chart in Fig. 8. The vapors from the central vapor column of the evaporator are taken out through a shell and connected into a header. The vapors pass from this header through an oil catcher similar to a steam separator, the condensate passing out of the bottom of it into the bottom of the evaporator. They then pass through six large dephlegmators in series, flowing into the bottom of each and out at the top. In each dephlegmator a partial condensing of the vapor takes place, thus forming a liquid in the bottom of that particular dephlegmator. From some of the dephlegmators this condensed liquid is a ready product; from others it lies just between two different products.

A system of water circulation is used as a cooling medium for the vapors from the dephlegmators and separators. The vapors are cooled in vertical tubular condensers, and about twenty barrels of water are required per barrel of distillate cooled. Superheated steam is used in the bottom of the evaporator, separators and dephlegmators as an agitator to relieve the lower boiler point fractions from the bottoms. About 30 pounds per barrel of distillate are produced.

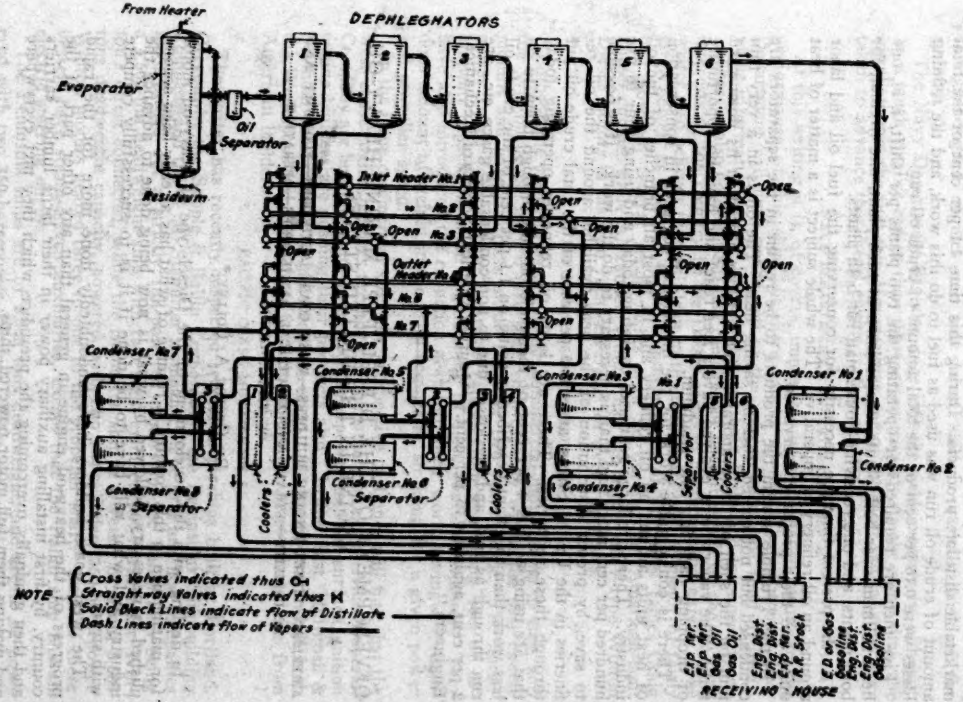

FIG. 8.—VAPOR AND DISTILLATE SYSTEM.

With two Trumble plants in operation continuously for eight months, the company for whom these plants were installed has averaged a run of 16,000 barrels of crude oil per day of 24 hours through both plants, 30 per cent of this crude oil being vaporized and fractionated into ready marketable distillate products. During this time 1.1 per cent of the total amount of crude oil run was used as fuel to do this work, and the refining losses were 0.75 per cent of the total amount of crude oil run.

Five men are required for operating the two plants per shift, *i. e.*, one head stillman over both plants, one stillman at each plant, one fireman for both plants, and one receiving-house man for both plants.

At the present time the necessity for conserving the fuel oil and labor around an oil refinery is imperative. The whole subject is a matter of heat units and their proper conservation.

In the Trumble system the products to be re-run in the separators are condensed but not cooled, thereby saving heat. The oil is in a very thin film on the shell of the evaporator and is being heated in its downward flow. The vapors being evolved have a free relief and have no opportunity of dropping back into the residuum.

There is considerable work for the mechanical engineer in conservation of heat units around an oil refinery and in most all branches of the oil industry. There are large refineries operating today and using four to five hundred per cent more fuel than is necessary to do the work. It is possible to save, by proper distribution of the heat available around the oil refineries in the United States, at least 2 per cent of the total crude oil put through these refineries. There are large oil refineries operating today that are using as fuel 4 per cent of the crude oil run through them to do less work than the plant described is doing on 1.1 per cent of the crude oil run through as fuel, and the losses due to non-condensable gases, etc., are 2 per cent, against 0.75 per cent.—"Journal American Society of Mechanical Engineers."

DEVELOPMENT OF THE DIESEL TYPE MARINE HEAVY-OIL ENGINE IN THE UNITED STATES.

AMERICAN DIESEL ENGINE BUILDERS—WEST COAST ACTIVITIES—ECONOMY AND RELIABILITY—STEAM AND ELECTRICAL AUXILIARIES—OPERATION OF THE ENGINES.

By George A. Colley.*

In dealing with the development of the Diesel type of heavy-oil motor for marine use in this country, a resumé of what has already been accomplished is necessary, followed by what is now being done to advance the industry and what will have to be done if it is to successfully compete with steam.

The west coast builders have undoubtedly done more for the rapid progress of the heavy-oil engine in general than any other part of the country, by first installing auxiliary power in their large lumber carriers and then gradually overcoming the prejudice which they met everywhere and making them full motor powered ships.

*Tams, Lemoine and Crane, Naval Architects, New York.

FOREIGN DESIGNS ADOPTED.

Spurred on by the sight of foreign-owned and built ocean-going motorships in our ports, a few of our large eastern yards have made arrangements for building Diesel engines from foreign designs, while several other foreign-designed engines are now being built in various parts of the country.

The Navy Department has also gone into the field extensively, although to a much smaller extent than the foreign navies. At the same time, the installation in the fuel ship *Maumee* was one of the most radical steps ever taken to advance the Diesel engine in this country.

NAVY DEPARTMENT'S INVESTIGATIONS.

At this writing it is, of course, impossible to give detailed results of the performance of this ship, but there can be no doubt but that our Navy Department will have at the end of the war an enormous fund of invaluable information on the Diesel engine of both high- and low-speed types.

There is also no doubt but that the close of hostilities will see the yachting interests of the country in the Diesel game to an increasing extent. The writer also believes they will lean to the full Diesel type of motor of fairly large power, as previous to the commencement of hostilities James Craig had powered several small yachts with an out-and-out American-built Diesel engine, while Alexander Winton has had a pair of V-type Winton Diesel motors in his own yacht *La Belle* for over two years.

REDUCTION GEARS INTRODUCED.

Both of the above firms are in the commercial motor game to stay, and the Winton people have tripled their shop capacity within the past twelve months and have contracts ahead for over two years' work. The horsepower called for in these contracts ranges from generating sets of 60-kilowatt capacity to main engine units of 1,000 shaft horsepower. They have also introduced the reduction gear into their work, saving a tremendous amount of weight in the main engines by so doing for a given power and propeller speed.

In the large commercial sizes the prospective customer will find that the west coast engine builders are alive to the situation, as the Atlas-Imperial Company are building units up to 1,200 horsepower, and the surface ignition engine builders are pushing their products also. In the East, McIntosh & Seymour are prepared to build engines up to 1,350 horespower, while both the Werkspoor and Burmeister & Wain motors can be contracted for from American builders.

PACIFIC COAST MOTORSHIPS.

The writer has recently inspected several motorships built at Portland, Oregon, by the Supple & Ballin Corporation, the Peninsular Shipbuilding Company and the G. M. Standifer Company. The first and last companies have turned out a strictly full-powered motorship, the Ballin ship being a 4,500-ton deadweight ship with two eight-cylinder, 500-horsepower Winton's direct connected, while the Standifer ship is a vessel of about 3,500 tons deadweight capacity, powered with two six-cylinder, 350-horsepower Winton motors geared three to one to the propeller shaft. Both of these vessels have steam-driven auxiliaries, except that the Ballin ship has an electric steering engine.

The Peninsula ships are of 2,300 tons deadweight, and are to be powered with two 350-horsepower Wintons, direct connected. They also are to have steam auxiliaries.

These vessels are representative types of what is being built throughout the Douglas fir district, and the workmanship throughout is of the best.

The various oil companies can be counted on to help things along, as they all have tried out various types of motor-driven ships with sufficiently satisfactory results to warrant further operations along the same lines.

INITIAL ERRORS MUST BE CORRECTED.

Mistakes have been made in the past which have retarded progress and created a distrust of the oil engine generally which will be hard to overcome, but a frank admission of errors and a willingness to correct them by builders both of hull and machinery will do much to help matters.

These mistakes and their remedies, also the various other points bearing on the building of a substantial foundation under the American motorship trade, the writer proposes to discuss in their relation to the various interested parties, namely, the owner or operating company, the operating engineer, the engine builder, the installing yard and the naval architect.

The owner, or operating company, has certain conditions which must be fulfilled. His requirements are that his installation shall be reliable, economical, both as regards running expense and upkeep, easy of repair and of reasonable first cost. He also must be assured that his auxiliaries will be taken care of properly.

RELIABILITY THE FIRST CONSIDERATION.

Regarding reliability, the average owner does not care to experiment at his own expense, and the engine builder should be prepared to meet him more than half way in regard to terms, guarantees, etc., and once having done so, to follow them up with the same amount of zeal that was used in obtaining the order.

The owner will also want to know if the particular size of machine he intends using has ever been built before. At present the average answer will be negative, as the average builder in this country has not had time to experiment on a machine of each of the various sizes he intends building. One builder has adopted a practice of building a single-cylinder machine of each size and obtaining data therefrom. This is a move in the right direction, as there has been too great a tendency toward selling machines regardless of their suitability for the work.

FUEL ECONOMY.

As regards fuel economy, the Diesel type has a tremendous margin over the best steam practice, but in many installations the lubricating-oil cost reduces this margin to a very large extent. A carefully worked out system of filtration and cooling for re-use of the lubricating oil will help materially.

Upkeep and ease of repair are vital. The repair bills and layoffs due to engine trouble must be reduced by careful attention to detail. Ease of repair can be furthered to a great extent by careful attention to the location of accessories and auxiliaries, as half of the repair expenses in many cases are due to the time taken to remove and replace piping and other gear in no way connected with the trouble, but so in the way that nothing can be done with them in place.

Regarding reasonable first cost based on power delivered, fair competition will probably take care of this question automatically.

AUXILIARIES—ELECTRIC INSTALLATION IDEAL.

The question of auxiliaries is one which many of the engine builders have overlooked to a large extent. Two general schemes present themselves for consideration, and the Diesel engine salesman should be well posted on both before attempting to talk business. They are, first, the straight steam-driven auxiliary installation supplied with steam from a donkey boiler,

which should be a real oil burner, not a makeshift, and, second, the electric-driven installation with oil engine-driven generating sets. Probably the electric installation properly worked up with breakdown sets for emergency use and a small water-tube, oil-fired boiler for heating, hot-water service, etc., would be the ideal installation.

Above all things in the auxiliary proposition, give the owner enough power. Many of the smaller motor ships at present are using the old-fashioned inefficient fire-tube donkey boilers of just capacity enough for the winches when the latter are new. After they have been in commission a short time there is a continual loss of time when loading and unloading, which, under present conditions, is nothing short of criminal.

STEAM AUXILIARIES.

A ship equipped with steam auxiliaries should be fitted with condensing and evaporating apparatus for make up. The writer also believes that a carefully worked out system of fresh-water cooling for the main engine would be well worth a trial.

Proper attention should also be paid to adequate bilge and fire service, electric lighting, etc. Possibly the engine builder will think these matters out of his province; but they are the questions which the owner will ask, and if he is met by answers that indicate the engineer's knowledge of his subject much will be gained.

The operating engineer is more properly interested in the improvement of fuel-handling devices, attention to lubrication problems, and the like. The writer believes that most of the builders will do well to bring into use many of the standard strainers, filters, coolers and the like, now on the market for fuel and lubricating oils, rather than to attempt to go all over the work of the very competent engineers who designed them. The writer has seen filters and strainers for use with Diesel motors which are crudities compared with the accessories used in steam practice.

TRAINED OPERATING ENGINEERS BADLY NEEDED.

Efficient operating engineers are at a premium and oil-engine manufacturers generally would do well to take a leaf from the books of some of the largest and most successful machinery manufacturers. These latter have schools for their embryo salesmen where they learn every detail of the machinery right in the shops, so that when sent on the road they are capable of selling, erecting and making the initial run of the machine. Some such scheme used with regard to engine crews would soon give us a set of men who would be boosting the game and not knocking it.

The engine builder has some very strong opinions with regard to owners and operating engineers generally, not without a good deal of cause. Many owners seem to think that because their machinery is oil-driven almost any sort of an engineer can run it, when, as a matter of fact, at this stage of the business it will take the closest kind of coöperation between builder and engineer to make things run smoothly.

BUILDERS SHOULD DO MORE THAN SUPPLY THE ENGINES.

The builders will also do well to follow the methods of the large marine-turbine manufacturers, who are always at the disposal of their customers for information of all kinds having a bearing on their installations. They are never content merely to sell their main units, but always make it a point to know whether the rest of the equipment, and even the ship itself, will co-ordinate with what they are furnishing to make a perfect whole.

The installing engineer, or shipyard, also has an extremely important part in the success of the outfit, more particularly so with regard to a full

Diesel installation than with the surface-ignition type. The high-pressure air lines must be installed with the greatest care, which means constant watchfulness in these days of green workmen and rush jobs. These lines should be as short as possible and the whole layout should be as simple as conditions will allow. This same thing applies to the entire main engine and auxiliary installation.

WHAT THE NAVAL ARCHITECT CAN DO.

The naval architect's connection with the advancement of the Diesel engine should be along the lines of the careful and accurate co-ordination of all the before-mentioned points. It should be his duty to impress on engine builders the necessity of obtaining proper engine speed for any particular type of ship. Underpowering and lack of attention to auxiliaries or other details will do much to slow up progress.

In conclusion, no time was ever riper for the advancement of any industry than is the present time for the advancement of the motorship. With the New York State barge canal about to open for traffic, and, in fact, all the inland waterways and coastwise trade of the country in need of motor-driven craft of all kinds, experience along these smaller lines can be gathered to such an extent that when hostilities cease the motorship should have no difficulty in giving the steamship the race of its life.—"Marine Engineering."

TOSI MARINE STEAM TURBINES.

(Abridged.)

In view of the widespread adoption of steam turbines for the propulsion of both naval and merchant vessels, readers of "Marine Engineering" will undoubtedly be interested in the developments which have been made by the firm of Franco Tosi, Legnano, Italy, the largest firm of steam turbine builders in Italy. For many years this firm has been engaged in the building of steam turbines, and have so far built turbines aggregating over 1,100,000 horsepower, in units varying from 12 horsepower up to 25,000 horsepower. Of these more than 800,000 horsepower have been for marine purposes. This firm today employs about 6,000 men in its shops.

The type of turbine adopted by this firm is shown in Fig. 1 and consists, as can be seen from the illustration, of one impulse wheel and one reaction drum, the latter constituting the intermediate-pressure and low-pressure stages of the turbine. If the turbine is built for a stationary plant it is built in one drum. If, on the other hand, the turbine is designed for naval work with reduction gears, it would be built in two drums, one for high pressure and the other for low pressure.

For low-powered turbines, or for turbines with medium or high power, but which have a high speed, if the intermediate stage were built complete as a reaction turbine the resulting efficiency would be too low because the first rows of blades would be too small; therefore, instead of using only one drum, this firm makes use of the two impulse wheels and one reaction drum. In marine turbines for direct drive, that is, coupled direct with the propeller shaft and, therefore, designed to work at relatively lower speeds, it would be of little advantage to adopt only a one-impulse wheel and one reaction drum for the intermediate and low-pressure stages, because the blades of the reaction drum, especially in the first rows, would be too small for efficient operation and, in consequence, the efficiency would be poor. For this reason, the Tosi firm builds turbines in which the impulse parts are increased and reaction parts diminished. This method has the advantage that the thrust which the steam produces at the beginning of the low-pressure stage of the turbine is directly equal to the propeller thrust.

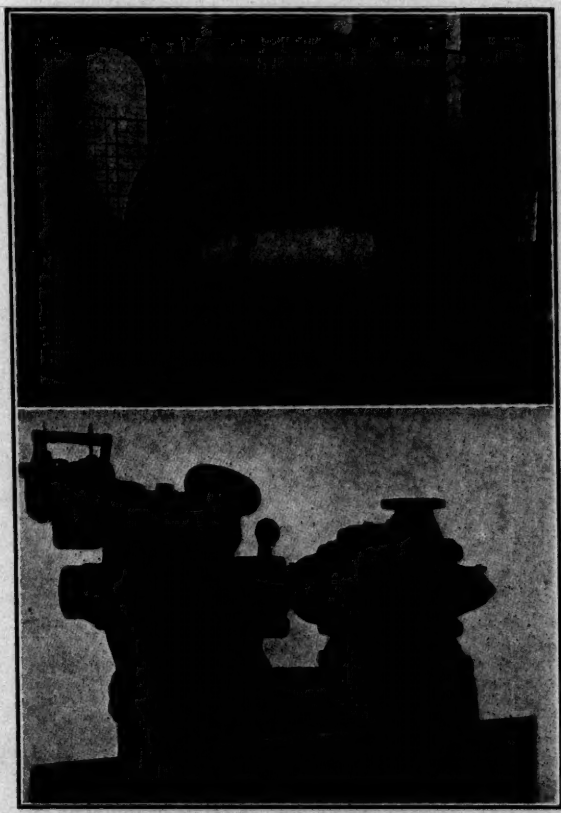

FIG. 1.—9,000–12,000 HORSEPOWER TURBINE, 750 REVOLUTIONS PER MINUTE.
FIG. 2.—TURBO BOILER FEEDING PUMP.

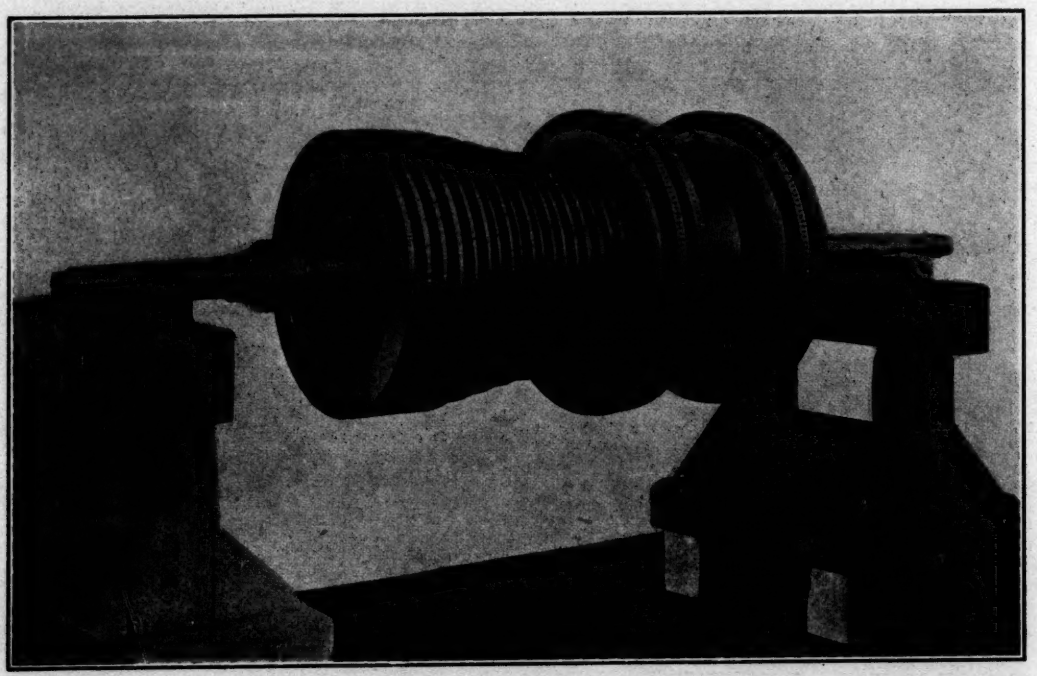

FIG. 3.—ROTOR OF 7,000-HORSEPOWER TURBINE.

FIG. 4.—20,000–25,000–HORSEPOWER MARINE TURBINE, DESIGNED TO RUN AT 500 REVOLUTIONS PER MINUTE.

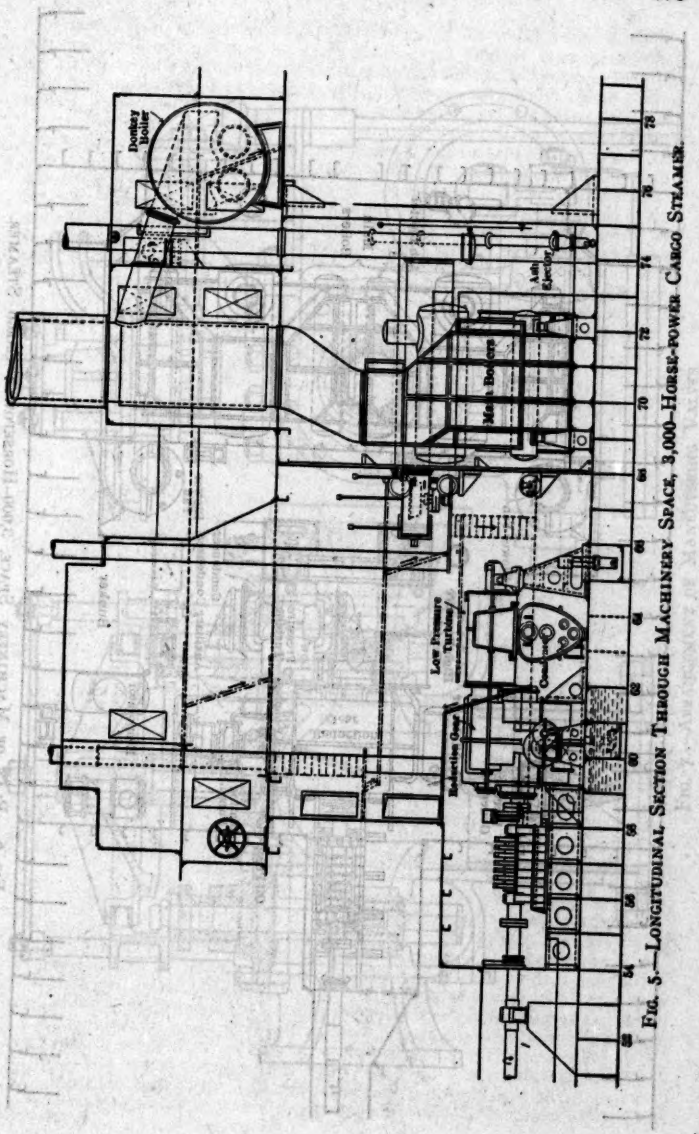

FIG. 5.—LONGITUDINAL SECTION THROUGH MACHINERY SPACE, 3,000-HORSE-POWER CARGO STEAMER.

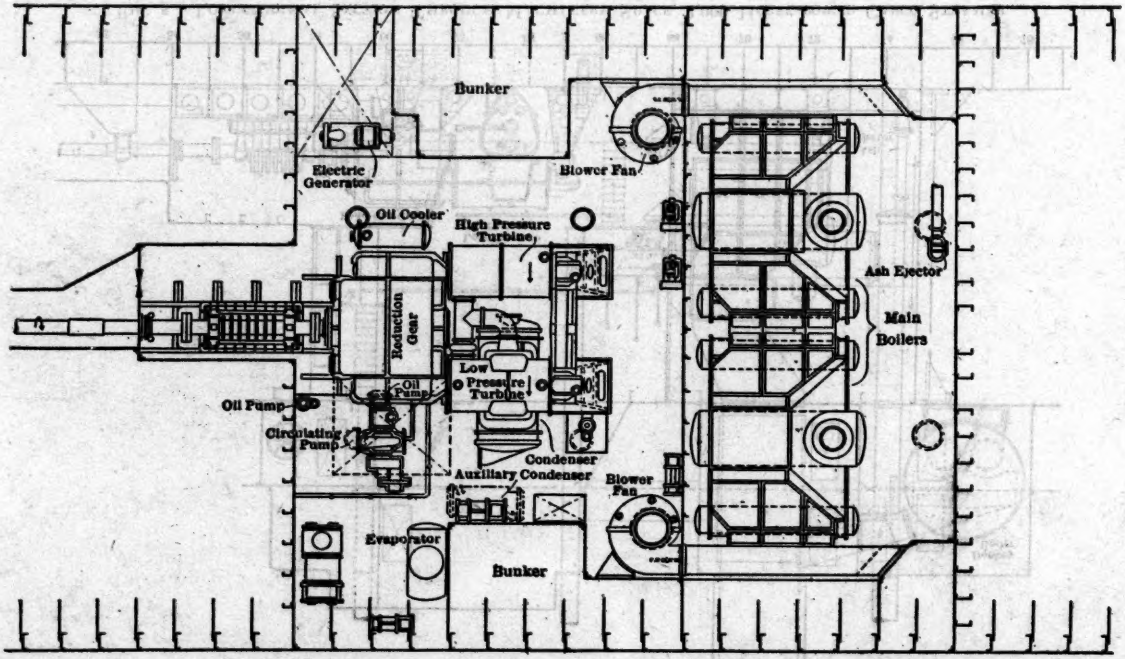

FIG. 6.—PLAN OF MACHINERY SPACE, 3,000-HORSEPOWER CARGO STEAMER.

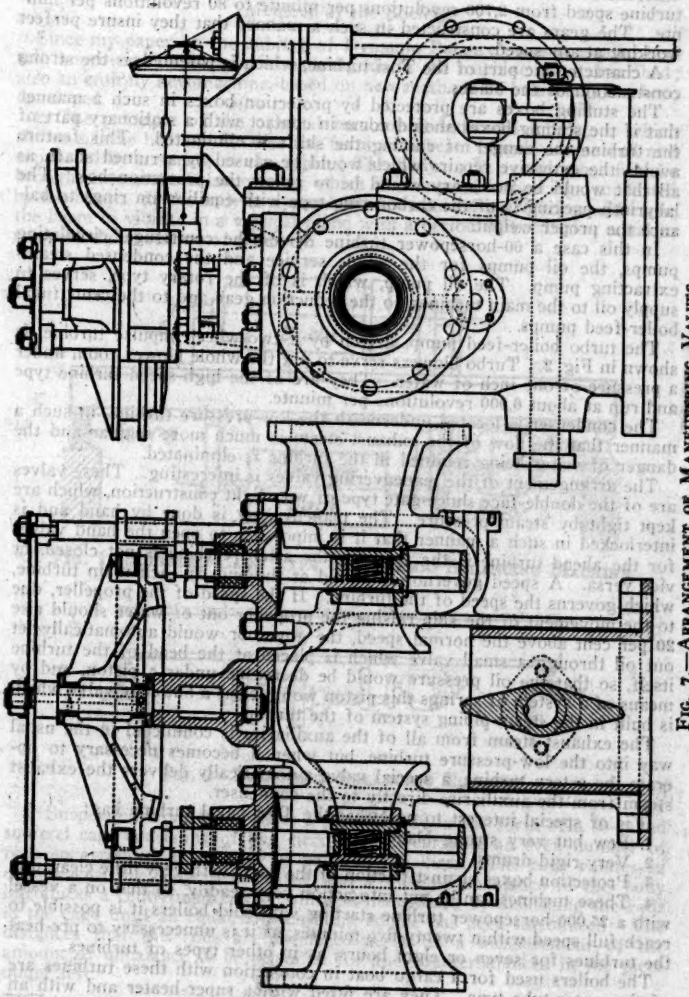

FIG. 7.—ARRANGEMENT OF MANEUVERING VALVES.

FIG. 6.—PLAN OF MACHINERY SPACE, 3,000-HORSEPOWER CARGO STEAMER.

Figs. 5 and 6 show the machinery arrangement for an installation of a 3,000-horsepower turbine for a cargo boat as built by the Tosi firm for different owners. Double reduction gears are provided which reduce the turbine speed from 2,700 revolutions per minute to 80 revolutions per minute. The gears are constructed in such a manner that they insure perfect working at any speed.

A characteristic part of the Tosi turbine, which is patented, is the strong construction of the blades.

The stuffing boxes are protected by protection boxes in such a manner that if the stuffing boxes should come in contact with a stationary part of the turbine the danger of ruining the shaft is eliminated. This feature avoids the extensiye repairs which would be caused by a ruined shaft, as all that would be necessary would be to renew the protection box. The labyrinth packing is of the carbon-ring type with equilibrium rings to balance the proper weight.

In this case a 60-horsepower turbine drives the centrifugal circulating pumps, the oil pumps for the whole service and the condensed steam-extracting pump. The oil pump, which is of the rotary type, serves to supply oil to the main turbines, to the reduction gear and to the centrifugal boiler-feed pumps.

The turbo boiler-feed pump, driven by a two-wheel impulse turbine, is shown in Fig. 2. Turbo blowers serve to put the whole blower room under a pressure of one inch of water. They are of the high-speed turbine type and run at about 6,000 revolutions per minute.

The condenser is located underneath the low-pressure turbine, in such a manner that the flow of the exhaust steam is much more regular and the danger of water being retained in the turbine is eliminated.

The arrangement of the maneuvering valves is interesting. These valves are of the double-face sluice-gate type of very light construction, which are kept tight by steam pressure. The maneuvering is done by hand and is interlocked in such a manner that it is impossible to open the hand valves for the ahead turbine if the valve of the astern turbine is not closed, or vice versa. A speed governor is placed at the end of each main turbine, which governs the speed of the turbine. If the speed of the propeller, due to the movement of the ship raising the propeller out of water should rise 20 per cent above the normal speed, the governor would automatically let out oil through a small valve which is placed at the head of the turbine itself, so that the oil pressure would be decreased under a piston, and by means of a system of springs this piston would close a butterfly valve which is built in the steam-piping system of the turbine.

The exhaust steam from all of the auxiliaries is conducted in the usual way into the low-pressure turbine, but when it becomes necessary to operate the astern turbine, a special valve automatically delivers the exhaust steam from the auxiliaries directly to the condenser.

It is of special interest to note that the Tosi naval turbine has:

1. Few but very strong blades.
2. Very rigid drums.
3. Protection boxes against friction of the parts with very little clearance.
4. These turbines can be put into action very readily, so that on a vessel with a 25,000-horsepower turbine starting with cold boilers it is possible to reach full speed within twenty-five minutes, as it is unnecessary to pre-heat the turbines for seven or eight hours, as in other types of turbines.

The boilers used for a cargo boat in connection with these turbines are of the water-tube type. They are fitted with a super-heater and with an arrangement to heat the air from the products of combustion before the air reaches the grates.—"Marine Engineering."

RECENT DEVELOPMENTS IN BALANCING APPARATUS.

By N. W. Akimoff, Philadelphia, Pa.,

Member of the Society.

Since my paper on the subject of Dynamic Balance* was presented certain improvements have been made by me in the machine there described; also an entirely new machine, based on new methods of balancing, has been developed.

2. The principle of the original machine is indicated in Fig. 1, which shows a lathe bed which takes the form of a beam hinged at one end and supported by a spring at the other. The body to be tested must first be brought into static balance, after which it is rotated in bearings supported by the beam. If the body is dynamically unbalanced its rotation will cause the beam to vibrate in a vertical plane with a period of oscillation equal to the period of rotation of the body.

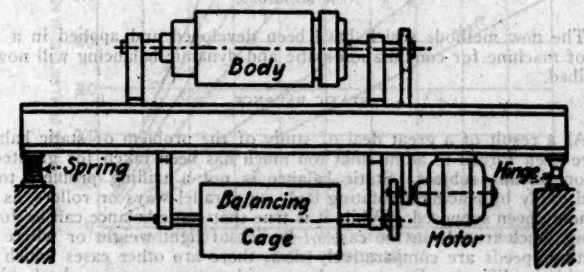

Fig. 1 Squirrel-Cage Type of Dynamic Balancing Machine

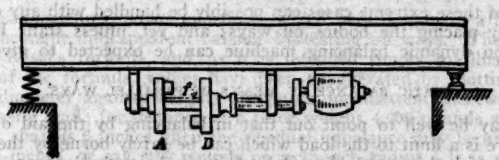

Fig. 2 Two-point Balancing Element

3. Suspended from the beam is a second body in the form of a so-called squirrel cage consisting of two circular discs carrying an even number of rods so arranged as to slide in holes in the discs. The cage rotates in unison with the body to be tested, and a state of unbalance in this body introduces a centrifugal couple which is neutralized by displacing the rods in the cage until an equal compensating couple has been introduced. The distances that the rods are displaced serve as a measurement of the amount of unbalance to be provided for and counterbalanced in the piece under test.

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* New Orleans Meeting, 1917; Trans. Am. Soc. M. E., vol. 28, p. 367.

Presented at the Annual Meeting, December 1917, of The American Society of Mechanical Engineers.

4. The improvements upon this machine, referred to above, were as follows:

a. The substitution for the cage of a two-point element consisting of two discs, *A* and *B*, each with a pin projecting from its face, as shown in Fig. 2. The disc *A* is fixed to its shaft and the disc *B* is arranged to slide on the shaft through the use of a feather key *f*. It is clear that when the two discs are in contact they will balance each other; but when separated they will introduce a certain centrifugal couple according to the weight of the pins and the distance between the discs, which latter can be varied while the apparatus is in motion.

b. A planetary arrangement by which the relative angular position of the body and the discs (or cage) can be varied while the machine is in operation.

c. The application of a principle whereby the discs (or cage) may be arranged to answer the problem of static balance as well as dynamic balance.

NEW METHODS.

5. The new methods which have been developed and applied in a new type of machine for combination static and dynamic balancing will now be described.

STATIC BALANCE.

6. As a result of a great deal of study of the problem of static balance I have been forced to admit that too much has been taken for granted in relation to this subject. Static balance is not a trifling problem to be solved easily by placing a rotating body on parallel ways or rollers, as has commonly been supposed. While it is true that static balance can be found without much trouble in the case of bodies of light weight or where the operating speeds are comparatively slow, there are other cases which are much more difficult. For example, consider a gyroscope wheel whose weight is about 50 pounds or so, running at, say, 10,000 r.p.m.; or a turbo-rotor whose weight is 10,000 pounds and whose speed is 3,600 r.p.m. Neither of these extreme cases can possibly be handled with any degree of success by placing the bodies on ways; and yet, unless static balance is perfect, no dynamic balancing machine can be expected to give reliable results.

STATIC BALANCE BY MEANS OF PARALLEL WAYS.

7. It may be well to point out that in balancing by the aid of parallel ways there is a limit to the load which can be safely borne by the journals in contact with the ways. A safe load for each journal appears to be 750 pounds per inch of width per inch diameter of journal.* For instance, if the ways are 1½ inches wide and the journal diameter is 10 inches, then each side will carry almost 12,000 pounds without any danger of forming permanent flat spots.

8. It would be of interest if one could estimate the sluggishness of action of a body on the ways under different conditions. The older theories of rolling friction, as proposed by Coulomb, Morin and Dupuit, do not seem to lead to very reliable results. Résal's formula† is probably much more reliable and is here reproduced in simplified form (steel on steel):

$$f = 0.056 \sqrt{1/(1+79/D)},$$

* According to the writer's remembrance, this figure was first given him by the engineers of the Standard Roller Bearing Co. many years ago, and has since been revised by him in connection with other practical data which he was able to gather from different sources.

† Résal, Traité de Mécanique Générale, T. 11, p. 6.

where f is the length of the flat contact of shaft with the way and D is the diameter of shaft, both in inches. It appears that the weight does not enter into the formula, and in general its results should not be considered as extending to extremes, but only as characterizing the average run of things as they are met with in practice.

9. The diagram in Fig. 3 has been roughly plotted to illustrate the prob-

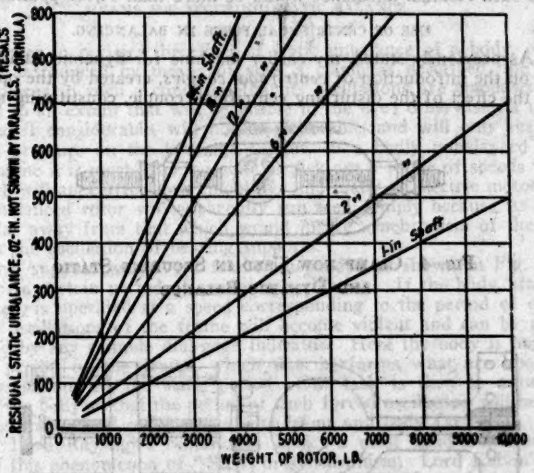

FIG. 3 DIAGRAM TO ILLUSTRATE PROBABLE RESIDUAL UNBALANCE OF BODIES BALANCED ON TESTING WAYS

able *sluggishness* or *residual unbalance* which *may* remain in a body that has been brought into apparent balance by testing on ways. The chart is based on Résal's formula, and, as it is intended simply to illustrate the meaning of the formula, values have been exaggerated by plotting shafts of small diameter against weights altogether inconsistent therewith. The chart is easily read: for instance, if a rotor weighs 4,000 pounds and diameter of its shaft is 6 inches, then the residual unbalance may be as great as 480 ounce-inches, or say, 2½ pounds on a 12-inch radius. It is useless to put a body statically unbalanced to such an extent (or even to 50 per cent of this) on any dynamic balancing machine. Satisfactory results cannot possibly be derived from such tests. An actual case of residual unbalance, typical of many others, is of a shaft having a diameter of 2⅛ inches, a weight of 88 pounds and a residual unbalance of 0.915 ounce-inches. The width of the balancing ways was 5/16 inch (cast iron chilled and ground).

10. Considering the phenomenon of rolling friction from the standpoint of higher theories of elasticity (Hertz), the following tentative formula was derived for residual unbalance:

$$M = 0.0004648\, P \sqrt{PD},$$

where M is the residual moment in ounce-inches, P the weight per unit of contact length (that is, per inch of combined width of ways), and D the diameter of shaft in inches (steel on steel). The constant may be considered to be rather tentative, but with the advent of a machine capable of establishing perfect static balance it will not take long to find more reliable values for it.

11. According to this formula, if an armature weighed 12,000 pounds and had a shaft diameter of 8 inches, the sluggishness or residual balance when placed on ways 1 inch wide, would be:

$$M = 0.0004648 \times 6{,}000 \sqrt{6{,}000 \times 8} = 614 \text{ oz-in.},$$

or over 3 pounds on a 12-inch radius. At 3,600 r.p.m. the centrifugal force due to such residual unbalance would be more than 14,000 pounds.

USE OF CENTRIFUGAL FORCE IN BALANCING.

12. As previously stated, my earlier machine for dynamic balancing was based on the introduction of *centrifugal couples*, created by the operator to offset the effect of the disturbing centrifugal couple, constituting unbalance

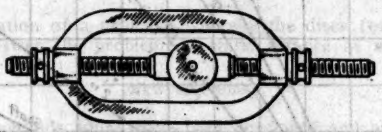

FIG. 4 CLAMP NOW USED IN SECURING STATIC AND DYNAMIC BALANCE

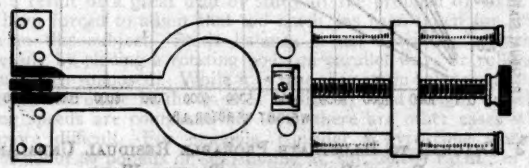

FIG. 5 A SECOND TYPE OF CLAMP

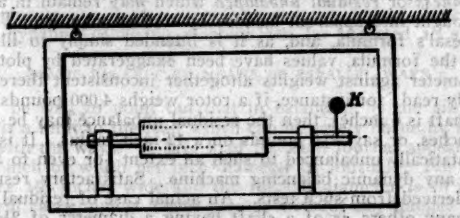

FIG. 6 BODY SUPPORTED IN SWINGING FRAME FOR SECURING STATIC BALANCE

in the rotating body. Likewise, a machine for static balancing can be based on the same principle, but long study of the subject has led me to the conclusion that the whole problem of balance, static and dynamic, can be reduced to the principle of a single *centrifugal force* acting on a properly constrained body. Such centrifugal force can be created by the operator, within a rotating body, by such means as, for instance, the *clamps* of which two designs are here shown (Figs. 4 and 5). As will be explained, such clamps may be used to offset the effect of static or dynamic unbalance in a rotating body, as the case may be, and to record the extent of such un-

balance. They are carefully made and so calibrated that the centrifugal force may be given as a function of some linear dimension, read directly or measured by an accurate scale. The first clamp is easier to make and check for accuracy, while the second design is much handier for quick adjustment on the shaft of a rotating body.

MEANS FOR SECURING STATIC BALANCE.

13. In order to register the effect of static unbalance of a body, or the correction introduced by means of such a clamp as described, the body must be placed in such a condition that its oscillations are emphasized or magnified to an extent that will be visible to the eye; otherwise its unbalance, even if considerable, will not be noticeable, and will only result in increased pressure on the bearings. Thus, in a badly unbalanced automobile engine it is most often possible to pick out a range of speeds where the engine will appear to work smoothly; and many an electric motor with badly unbalanced rotor will apparently run well, simply because its speed may be far away from that which would insure synchronism of the rotation with the oscillation of bearing supports.

14. Now, suppose we have a frame, suspended as shown in Fig. 6 and capable of a certain period of swinging oscillation. If the body, statically unbalanced, is operated at a speed corresponding to the period of oscillation, the oscillations of the frame will become violent and can be readily registered by any suitable dial-gage indicator. Here the body is imposing its own period on the frame, which thus performs what are known as forced vibrations of the same period. Our task is then to adjust the speed of the body so that the period of such forced oscillation will be equal to that of the natural oscillation of the frame and body (at rest).

15. In Lord Rayleigh's remarkable book* a most lucid explanation is given of this phenomenon of "Step" or synchronism. Lord Kelvin's well-known device called a "vibrometer" is likewise based on this very principle. Of course, it is easy enough to see that a clamp K can always be so adjusted, both angularly and as regards its off-center position, as to nullify the oscillations of the frame, thereby solving the problem of static balance. No matter how heavy the body, it is always possible to place it into most minute balance by this method, where ordinary parallels would be altogether inoperative.

DYNAMIC BALANCE.

16. As regards dynamic balance, due, in a statically balanced body, only to the presence of a centrifugal couple, the following remarks can be made: In the first place the "theory" that this centrifugal couple is due to the fact that the centers of gravity of both halves of the body, cut through its center of gravity, do not lie on the axis of rotation, is radically wrong. Take, for instance, a skeleton body shown in Fig. 7. Its center of gravity is exactly on the axis of rotation, as also are the individual centers of gravity of each half, to the right and to the left from A–B. Yet such a body would be manifestly out of balance (dynamically). Inversely, a body could readily be imagined to be both in static and dynamic balance, although each of its *halves* were statically out of balance. The only correct way to characterize dynamic balance is to say that the *products of inertia*, containing the axis of rotation, vanish;† or, to put it practically, that there is no centrifugal couple in any axial plane.

* Theory of Sound.
† Slocum, Theory and Practice of Mechanics, p. 297.

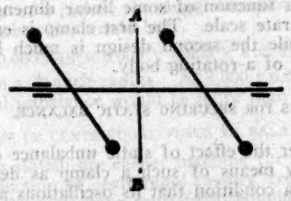

FIG. 7 UNBALANCED BODY WITH CENTER OF GRAVITY LYING IN ITS AXIS

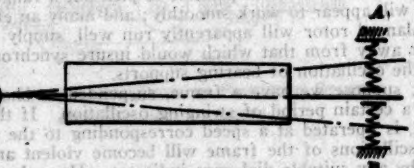

FIG. 8 ONE BEARING PIVOTED AND ONE FLOATING

FIG. 9 CENTRIFUGAL COUPLE BALANCED BY A CENTRIFUGAL FORCE

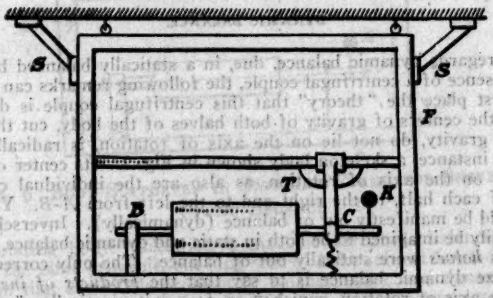

FIG. 10 BODY WITH ONE FLOATING BEARING, SUPPORTED IN RIGID FRAME FOR SECURING DYNAMIC BALANCE

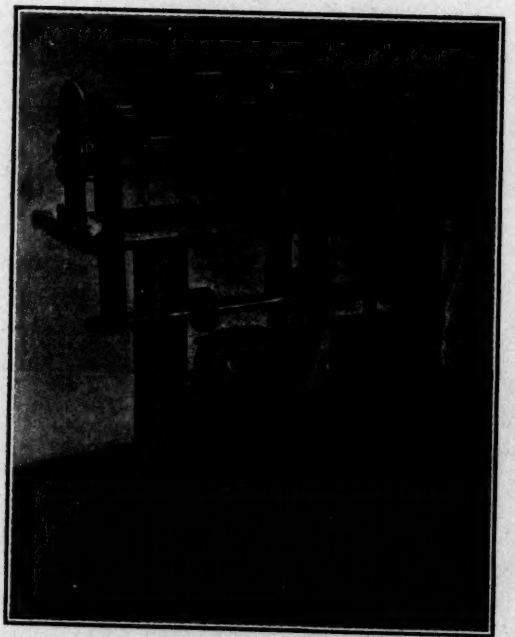

FIG. 11.—COMBINED STATIC AND DYNAMIC BALANCING MACHINE.

17. In the next place, if we constrain (pivot) one end of a rotating body (statically balanced but dynamically out of balance) while the other end is arranged to float in a bearing supported by springs so that it may move, say, in a horizontal plane, Fig. 8, then the oscillations of the body will be angular, as from *A* to *B*. Under these conditions the observer will be unable to tell whether the vibrations are due to a *force* (centrifugal) acting somewhere on the body or to a centrifugal couple, unless he knows beforehand that the body is in perfect static balance, under which condition the vibratory effect can be due only to dynamic unbalance. This being the case, in view of the reaction of the constrained end, it is perfectly possible to balance the effect of a centrifugal couple by means of a centrifugal *force*. Thus, in Fig. 9 if it is assumed that the dynamic unbalance is due to the couple *a–b*, it will always be possible to select a centrifugal force *c*, such that it will quiet the vibrating body, and because of its known distance from the bearing *B*, establish the exact value, sign and angular position of the disturbing centrifugal couple *a–b*.

18. It is thus clearly seen that it is possible to utilize a centrifugal force to good advantage in finding both static and dynamic unbalance of bodies; and combining the principles illustrated in Figs. 6 and 8, we have a combination static and dynamic balancing machine, of which the scheme is as follows:

DESCRIPTION OF COMBINATION BALANCING MACHINE.

19. A frame *F*, Fig. 10, supports the bearings *B* and *C*, which carry the body. The frame has a swinging period of its own. The bearing *C* may either be locked, so that it acts exactly like the rigid bearing *B*, or else it may be allowed to float in a vertical plane, bringing into play certain resistances (springs) opposing its deflection from the neutral (vertical) position. The correcting centrifugal force is indicated by *K*.

20. Such a system is known in dynamics as a *system with two degrees of freedom*, in general being capable of two kinds of motion: swinging of the frame, the bearing *C* being maintained rigid; and swinging of the bearing *C*, the frame *F* being maintained rigid by such means as brackets *S*; while the most general motion consists of a combination of these two motions.

21. The operation of such a combination machine is very clear. In order to secure static balance we lock the bearing *C* and unlock the frame supports *S*. Then, by properly adjusting the magnitude and direction of *K* we can reduce to zero the bodily oscillations of the frame *F*, thus establishing the exact value and sign of static unbalance in ounce-inch units. As soon as this has been corrected, we lock the frame *F* and unlock the bearings *C*, when the same centrifugal force *K*, created by a suitable adjustment of the clamp as explained above, can be made to correct the body for dynamic unbalance, as illustrated in Fig. 9. The advantage of basing the results on centrifugal force instead of on a centrifugal couple is manifest, the former being a *fundamental*, and the latter a *derived*, unit, so that the former is capable of much greater accuracy in adjustment and of more direct application than the latter.

22. Fig. 11 illustrates a balancing machine built in accordance with the scheme of Fig. 10. The yielding support, clearly shown on the right, has means for easy adjustment of its period, as likewise has the frame itself. The motor is of 1/3 H.P. capacity and operates the body, through a counter-shaft, by a rubber belt. The balancing clamp is seen on the extreme right of the crankshaft. The oscillations are read by means of ordinary Starrett dial gages, graduated in thousandths of an inch. The precision which can be secured on such a machine is almost uncanny; it

enables one to see the sluggishness of the method of balancing on ways and therefore the absolute lack of precision of dynamic balance that might be based on such results. A well-designed clamp is very easily handled and its correct position can be established in a few minutes. Its indications are capable of tabular interpretation, so that the operator has merely to carry out the simple instructions, worked out beforehand. Of course, it is clear that such a machine can be built for any size of body, or for any speed that may be desired.

APPLICATION OF ELECTRICITY TO VARIOUS AUXILIARIES ON SHIPBOARD.

By H. L. Hibbard.

Abstract.

In the use of electricity the American Navy has not only been far in advance of the merchant marine, but has led all navies of the world. Europe, however, seems to have led this country in the trial of electrical apparatus for auxiliaries on merchant vessels.

Among the most striking examples of the use of electricity on merchant vessels are the boats being constructed and projected by the Pennsylvania Shipbuilding Company at Gloucester City, N. J. This company has under construction six tank steamers of about 7,500 tons displacement, one of which is nearly completed and another will take the water shortly, which will be propelled by geared steam turbines and their auxiliaries operated by electricity throughout.

In addition to the above tankers, two cargo vessels of 12,000 tons displacement will soon be laid down and which will have an equipment of auxiliary motors similar to the above tank vessels with the exception of the deck winches, which will consist of three of 25 horsepower and eight of 15 horsepower. It is further contemplated that these two cargo vessels will be electrically propelled and operated by two turbo generating sets, each set furnishing current to a 1,500-horsepower motor, the two motors being geared to the single propeller shaft.

In addition to these vessels, this company contemplates the construction also of six cargo boats propelled by geared steam turbines and with electrical auxiliaries similar to the above.

For the above vessels open-type alternating-current motors will be provided throughout and watertight enclosing cases furnished for the deck motors, with the particular view to protecting them from exposure to moisture; but the cases can be removed, if desired, when the vessels are in port, to give the motors ample ventilation.

In contrast to the alternating current used on the above vessels, the Texas Oil Company is soon to construct a tank vessel of approximately 6,000 tons displacement, propelled by fuel-oil engines and provided with direct-current, 230-volt motors on its auxiliaries.

While it is not here the intention to discuss in detail the relative merits of various kinds of motors, it may be mentioned that in comparing the direct-current with the alternating-current equipments mentioned above, a saving in wiring, flexibility of speed control, and improved speed and torque characteristic for the deck auxiliaries are on the side of the direct-current equipments. The alternating-current motor, as pointed out above, has the advantage of freedom from moving contacts and the reduction, in consequence, to a minimum of the possibility of sparks in places where inflammable gases might be encountered, as in the case of the tank

steamers. This, however, applies particularly to the constant-speed, squirrel-cage type of alternating-current motors, the slip-ring type of motor, which is required for some applications being provided with moving contacts.

In connection with the electric propulsion of ships it is interesting to note that, in addition to the very considerable number of electrically-propelled ships now being constructed for the Navy, this mode of propulsion is receiving considerable attention in its application to merchant vessels. In addition to the two cargo vessels at the Pennsylvania Shipbuilding Company's plant, which it is expected to propel electrically, we are familiar with electrical installations on four oil barges under construction at New Orleans by the Petroleum Transport Company. These boats are twin screw, each propeller being driven by a 300-horsepower, 480-volt, 3-phase, 60-cycle motor, the two motors being operated by one turbo-generating set of similar voltage and frequency.

We have thus seen some of the progress which has been made in this application of power in the merchant field, and we feel certain that its application by this time would have been considerably more extended had not the war conditions necessitated, at least in the opinion of the Government officials, foregoing consideration of electrical apparatus in favor of steam in order to produce a merchant fleet in the shortest possible time.

In addition to the advantages which have been considered and resulted from electrical installations on naval vessels, numerous specific advantages are apparent in connection with such installations in the merchant marine, including frequently a saving in space and weight in the installation of the auxiliaries themselves, a very much higher efficiency of operation, and consequent saving in total fuel consumption, the elimination of long and objectional lines of steam piping, which on certain classes of cargo vessels are nearly always installed above deck and result in continual steam condensation; also, on certain of the auxiliaries, particularly those requiring varying speed and load conditions, the electric motor offers some distinct advantages over the use of the steam engine.—"Marine Engineering."

PROBLEMS OF CRANKSHAFT DESIGN.

A PAPER IN WHICH MATHEMATICAL DEDUCTIONS FROM EXAMPLES FROM PREVAILING PRACTICE ARE UTILIZED TO DERIVE FACTORS OF SAFETY AND OTHER SPECIFIC VALUES.

By Otto M. Burkhardt,* Buffalo, N. Y.

Gasoline engines, of the kind at present produced in large quantities for airplanes and motor vehicles, may turn over at 3,000 r.p.m. or faster. The forces necessary to induce and maintain these speeds, as well as other forces closely associated with high speeds, are numerous; but with a particular object in mind, this paper will be confined to the three most important groups of forces—the pressures due to the gaseous mixture, the inertia forces, and the centrifugal forces. The smooth running and the life of an engine depend mainly on these three factors.

The reciprocating masses linked to the crankshaft will be considered as one mass concentrated at one point in the axis of the cylinder. This simplification implies that the inertia forces are, like the gas pressures, acting primarily in the direction of the cylinder axis, a condition that permits

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arithmetical addition of both groups of forces. To the same category belongs another group of forces which, according to a well-known law, has its origin in the angularity of the connecting rod. The different component forces have been determined in respect to two engines of equal capacity for 24 crank positions. These positions are uniformly spaced at intervals of 30 degrees and comprise two revolutions, which constitute one complete cycle in case of four-stroke-cycle engines.

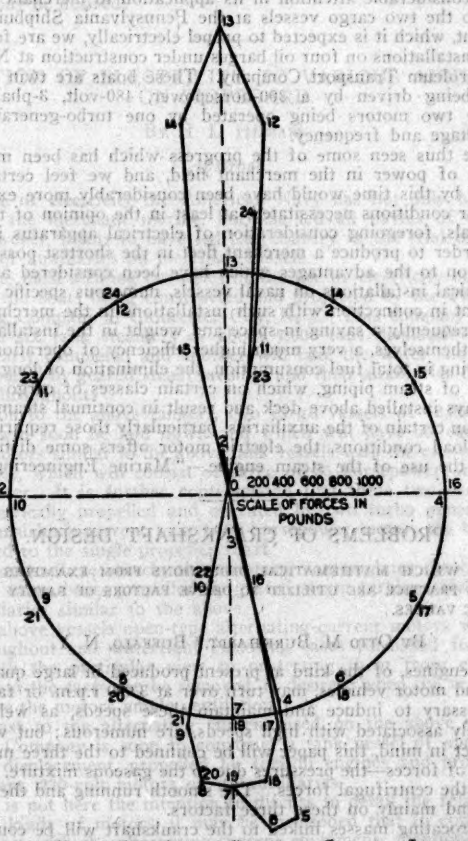


FIG. 1.—CENTRIFUGAL FORCES AND COMBINED GAS AND INERTIA FORCES ACTING ON CRANKPIN OF SIX-CYLINDER ENGINE.

TABLE 1.—DIMENSIONS AND WEIGHTS FOR ENGINES OF SIMILAR DESIGN.

	Six-Cylinder.	Twelve-Cylinder.
Cylinder bore, inches	3.75	3.00
Stroke, in. (=bore × 1.667)	6.25	5.00
Weight of reciprocating parts, lb. (one cylinder)	4.07	2.60
Weight of rotating parts, lb. (one cylinder)	2.40	1.93
Weight of rotating parts, lb. (two cylinders)	....	3.86

Corresponding components when combined as resultant forces and graphically represented in magnitude and direction, yield irregular characteristic diagrams with which every engine designer should acquaint himself.

As subjects for investigation, medium-size six- and twelve-cylinder engines, both of the same cylinder volume, were chosen. Both engines are supposed to be similar in design and up to the same standard of construction. Some particulars relating to these engines are given in Table 1.

Fig. 1 is a force diagram pertaining to the six-cylinder engine. The concentric circle is a graphical representation of the centrifugal forces acting on the crankpin at a speed of 2,700 r.p.m. The combined gas and inertia forces as above referred to are represented by the irregular polygons.

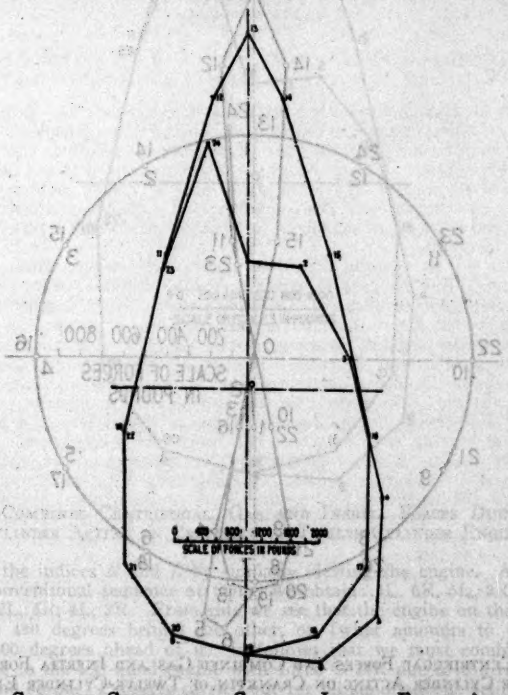


FIG. 2.—COMBINED CENTRIFUGAL, GAS AND INERTIA FORCES ACTING ON CRANKPIN OF SIX-CYLINDER ENGINE.

For instance, O-3 of this diagram represents the magnitude and direction of the combined gas and inertia forces if the crankpin is in position 3, that is, 60 degrees from the top dead center. For one complete cycle the different strokes are represented as follows:

Power stroke, by forces O-1 to O-7
Exhaust stroke, by O-7 to O-13
Suction stroke, by O-13 to O-19
Compression stroke, by O-19 to O-1.

To obtain a clear picture of the total forces acting on a crankpin, we must combine both diagrams as follows: O-1 of the polygon with O-1 of the concentric-circle diagram, and so on. This yields the diagram shown in Fig. 2, which represents the magnitude and direction of the resultant forces acting on the crankpin of a six-cylinder vertical engine.

The combined gas and inertia forces for one unit of a twelve-cylinder engine and the corresponding centrifugal forces are diagrammatically represented in Fig. 3. We observe that these forces are much smaller but quite analogous to those given in Fig. 1. The combination of the gas and

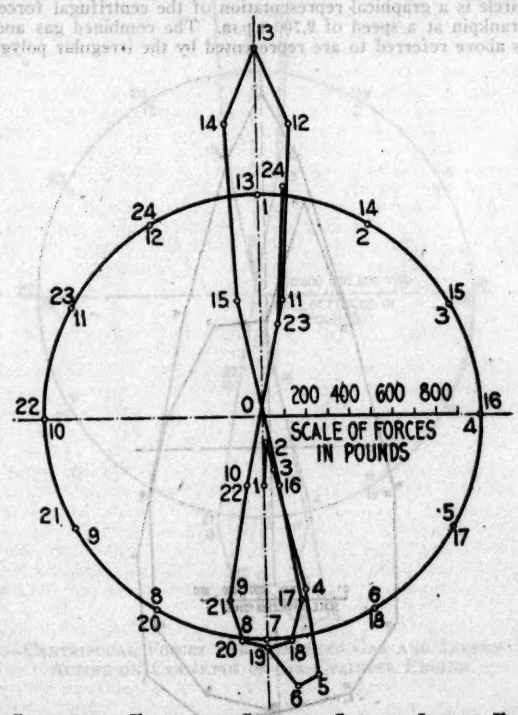


FIG. 3.—CENTRIFUGAL FORCES AND COMBINED GAS AND INERTIA FORCES DUE TO ONE CYLINDER ACTING ON CRANKPIN OF TWELVE-CYLINDER ENGINE.

inertia forces with the centrifugal forces yields the diagram shown in Fig. 4, which represents in magnitude and direction the resultant forces per cylinder acting on the crankpin of a twelve-cylinder engine. It must be borne in mind, however, that in the case of the engine under consideration, two single-cylinder engines are acting on one crankpin. We have, therefore, to superimpose two diagrams so that their vertical axes include an angle of 60 degrees, as in Fig. 5. The crank positions 1 to 24 in the diagram are plotted according to the clockwise rotation of the crankshaft. The distinction between the right- and left-hand block of cylinders is made

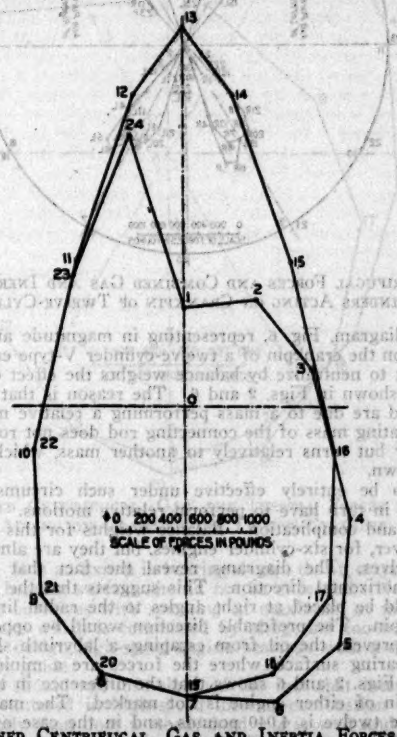


FIG. 4.—COMBINED CENTRIFUGAL, GAS AND INERTIA FORCES DUE TO ONE CYLINDER ACTING ON CRANKPIN OF TWELVE-CYLINDER ENGINE.

through the indices *R* and *L* by similarly viewing the engine. According to the conventional sequence of firing, we obtain: 1L, 6R, 5L, 2R, 3L, 4R, 6L, 1R, 2L, 5R, 4L, 3R. From this we see that the engine on the right is in phase 420 degrees behind the other, or (what amounts to the same thing) 300 degrees ahead of it. It follows that we must combine force O-1L due to the left-hand engine with force O-11R due to the right-hand engine and so on. Properly carried out for all simultaneous acting forces,

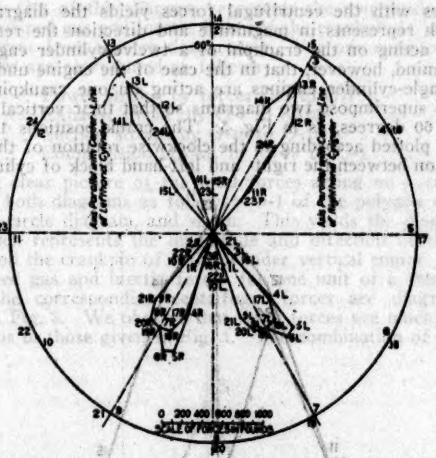


FIG. 5.—CENTRIFUGAL FORCES AND COMBINED GAS AND INERTIA FORCES DUE TO TWO CYLINDERS ACTING ON CRANKPIN OF TWELVE-CYLINDER ENGINE.

this yields a diagram, Fig. 6, representing in magnitude and direction the forces acting on the crankpin of a twelve-cylinder V-type engine.

It is difficult to neutralize by balance weights the effect on the crankpin of the forces shown in Figs. 2 and 6. The reason is that the centrifugal forces involved are due to a mass performing a relative motion. That is to say, the rotating mass of the connecting rod does not rotate truly about its own center but turns relatively to another mass, which turns about a center of its own.

In order to be entirely effective under such circumstances, balance weights would in turn have to perform relative motions. This introduces extra friction and complication. Balance weights for this effect are quite feasible, however, for six-cylinder engines, but they are almost an impossibility for twelves. The diagrams reveal the fact that the forces are smallest in a horizontal direction. This suggests that the oil holes in the crankpin should be placed at right angles to the radial lines through the center of the pin. The preferable direction would be opposite to that of rotation. To prevent the oil from escaping, a labyrinth should be cut in the bushing bearing surface, where the forces are a minimum.

A glance at Figs. 2 and 6 shows that the difference in the loads acting on the crankpin of either engine is not marked. The maximum load in the case of the twelve is 4,040 pounds, and in the case of the six, 4,795 pounds. The difference between the two is 18¾ per cent of the former. These loads increase approximately as the square of the speed. If then the twelve-cylinder engine runs at 2,940 r.p.m., it is subject to the same maximum load as is the six when running at 2,700 r.p.m. Mention is made of this because it is generally found in practice that a car with a twelve-cylinder engine is geared so that the engine runs at a slightly higher speed than a six-cylinder engine would be made to run.

The life of a crankshaft bearing depends largely on the magnitude of the mean pressure resulting from the various loads acting during one complete cycle, and this has been found to be 15 per cent smaller with the six than

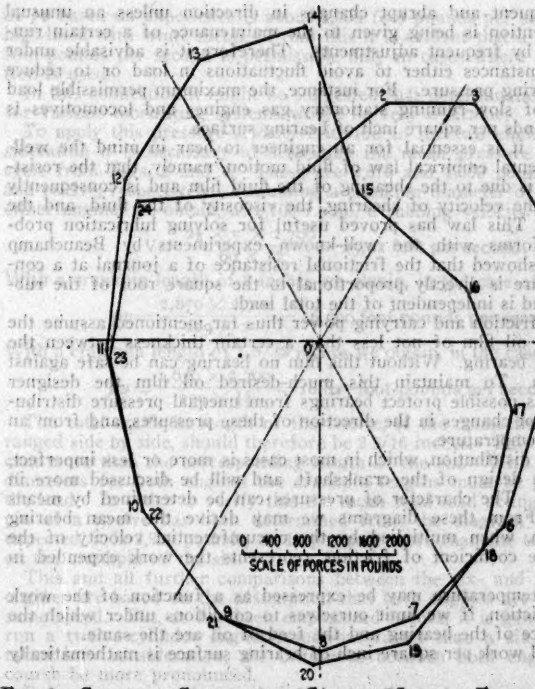


FIG. 6.—COMBINED CENTRIFUGAL, GAS AND INERTIA FORCES DUE TO TWO CYLINDERS ACTING ON CRANKPIN OF TWELVE-CYLINDER ENGINE.

with the twelve. It is an axiom in bearing design that a certain permissible load per unit of bearing surface should not be exceeded. Nevertheless we shall not content ourselves with speaking of maximum or mean loads exclusively, a mistake which is committed by altogether too many engineers who endeavor to design high-speed-engine bearings with a certain specific load as their only guide.

Designers also should not lose sight of the fact that intermittent loads such as are encountered in high-speed internal-combustion engines affect bearings differently than does a steady load. It is well understood that a load acting continually in one direction is likely to cause lubrication difficulties. To what extent a bearing will behave well under the effect of a steady load is shown by an experiment* made by Professor Goodman, who states that he has had a journal running for weeks with a surface velocity of 4 feet per second under a steady load of 2 tons (4,480 pounds) per square inch, the journal being kept at a temperature of 110 degrees F. by a stream of water forced through it. It is evident that bearing loads of this magnitude are hardly permissible under conditions where the load is

*Unwin, Elements of Machine Design, part 1, p. 243.

subject to frequent and abrupt changes in direction unless an unusual amount of attention is being given to the maintenance of a certain running clearance by frequent adjustments. Therefore it is advisable under ordinary circumstances either to avoid fluctuations in load or to reduce the specific bearing pressure. For instance, the maximum permissible load on crankpins of slow-running stationary gas engines and locomotives is about 1,500 pounds per square inch of bearing surface.

Furthermore, it is essential for an engineer to bear in mind the well-known fundamental empirical law of fluid motion, namely, that the resistance to sliding is due to the shearing of the fluid film and is consequently a function of the velocity of shearing, the viscosity of the fluid, and the shearing area. This law has proved useful for solving lubrication problems and conforms with the well-known experiments by Beauchamp Tower† which showed that the frictional resistance of a journal at a constant temperature is directly proportional to the square root of the rubbing velocity and is independent of the total load.

The laws of friction and carrying power thus far mentioned assume the presence of an oil film of not less than a certain thickness between the journal and the bearing. Without this film no bearing can be safe against undue abrasion. To maintain this much-desired oil film the designer should as far as possible protect bearings from unequal pressure distribution, from abrupt changes in the direction of these pressures, and from an undue rise in temperature.

The pressure distribution, which in most cases is more or less imperfect, depends on the design of the crankshaft, and will be discussed more in particular later. The character of pressures can be determined by means of diagrams. From these diagrams we may derive the mean bearing pressure, which, when multiplied by the circumferential velocity of the journal and the coefficient of friction, represents the work expended in friction.

The rise in temperature may be expressed as a function of the work expended in friction, if we limit ourselves to conditions under which the radiating surface of the bearing and the feed of oil are the same.

The frictional work per square inch of bearing surface is mathematically expressed by:

$$w = \mu \frac{P}{dL} \times \frac{d\pi N}{12 \times 60} = \mu p v \qquad (1)$$

where

w = work expended in friction per second and per square inch of bearing surface
μ = coefficient of friction
P = total bearing load, pounds
p = specific bearing load, pounds per square inch
d = diameter of shaft, inch
L = length of bearing, inch
N = speed of shaft, revolutions per minute
v = circumferential velocity of shaft, feet per second

From [1] it follows that

$$pv = \frac{PN}{229L}. \qquad (2)$$

According to data gathered from well-performing and durable engines, it may be stated that the value pv should not exceed 17,000 foot-pounds per second. This value is far in excess of other similar values. Güldner states that in case of stationary gas engines the maximum limiting value

† Proc. Inst. M. E., 1883 and 1884.

of pv based on experience is about 1,500 foot-pounds per second for bearings lined with white metal. From this it is evident that a value of 17,000 foot-pounds per second is permissible only in cases where forced lubrication is used.

In this connection it might be suggested that determination of pv values for various lubricating systems would be very desirable.

To apply this pressure-velocity criterion to the two engines under consideration it is necessary to determine the mean pressure of a complete cycle from Figs. 2 and 4. These are 2,870 and 1,650 pounds for the six and twelve, respectively. Assuming a crankpin diameter of 2⅛ inches for either engine, we obtain, at 2,700 r.p.m., a rubbing velocity of

$$V = \frac{2\frac{1}{8} \times \pi \times 2{,}700}{60 \times 12} = 25.02 \text{ feet per second.}$$

With a crankpin length of 2 inches we obtain, in the case of the six,

$$pv = \frac{2{,}870 \times 2{,}700}{229 \times 2} = 16{,}900 \text{ foot-pounds per second.}$$

With a crankpin length of 1 5/32 inches we obtain, in the case of the twelve,

$$pv = \frac{1{,}650 \times 2{,}700}{229 \times 1\frac{5}{32}} = 16{,}825 \text{ foot-pounds per second.}$$

The total crankpin length of the twelve, if the connecting rods are arranged side by side, should therefore be 2 5/16 inches. This is about 15 per cent more than the corresponding length of the six. With equal crankpin diameters for both engines the crankpin lengths must be proportional to the load. In order to verify this the mean total load acting on the crankpin of a twelve has been determined from the diagram in Fig. 6. As expected, it is 3,300 pounds, or 15 per cent more than the total load acting on the crankpin of the six.

This and all further comparisons between the six- and twelve-cylinder engines are based on the assumption that both engines are running at the same speed. But, as has been emphasized before, it is general practice to run a twelve-cylinder engine somewhat faster than a six. If this were taken into consideration, the difference between both engines would of course be more pronounced.

Before accepting the crankpin lengths given above as representing the final values, we will ease our minds with regard to the magnitude of the specific bearing pressure. In the case of the six we have a maximum load of 4,795 pounds. The projected bearing area of a crankpin 2⅛ inches in diameter and 2 inches long is 4¼ square inches. This gives a specific pressure of 1,125 pounds.

In the case of the twelve we have a maximum load for each cylinder (taken from Fig. 4) of 2,660 pounds and a projected bearing area of $2\frac{1}{8} \times 1\ 5/32 = 2.46$ square inches. This necessitates a specific bearing pressure of 1,080 pounds. The difference between both values is only 4.1 per cent, and both are of such magnitude as to be just about permissible. If the engines are to be used for automobile propulsion, then it may be borne in mind that maximum speed is generally not maintained for any great length of time. The loads above given would represent, then, the exception and not the rule.

Automobile engines, however, are frequently subject to hard usage when long hills are climbed on high gear. Under such conditions the explosive pressures alone determine the loads on the bearings, because the speed of the engine is generally so low that the inertia and centrifugal forces are negligible. Assuming an explosion pressure of 380 pounds per square inch for both engines, we obtain total explosive loads of 4,200 and 2,690 pounds for the six and the twelve, respectively. In the first case we obtain a

specific crankpin pressure of 4,200/4.25 = 990 pounds, while in the second we obtain 2,690/2.46 = 1,095 pounds. The difference between both engines is here 10.6 per cent in favor of the six. The loads of the two engines due to explosive pressure alone are somewhat smaller than those obtained before at high engine speed.

All forces acting on the crankpin necessitate reactions on the main bearings. The total load on the main bearings also includes the centrifugal forces due to the crankpin and the adjacent crank cheeks. Theoretically, and now even practically, it is a simple matter to attach weights to the crankshaft that will relieve the main bearings on the centrifugal forces. Whether it is an advantage to do so will be presently investigated by means of diagrams.

The forces to be dealt with at first are those relating to the six-cylinder engine resolved into components in a manner as determined for a crankshaft of the seven-bearing type.

We are now concerned with two groups of diagrams:

a Main-bearing load for a crankshaft without balance weights: Fig. 7, loads on main bearings I and VII (counting from front of engine); Fig. 8, loads on II, III, V and VI; and Fig. 9, load on IV.

b Main-bearing load for a crankshaft with balance weights: Fig. 10, loads on main bearings I and VII; Fig. 11, loads II, III, V and VI; and Fig. 12, load on IV.

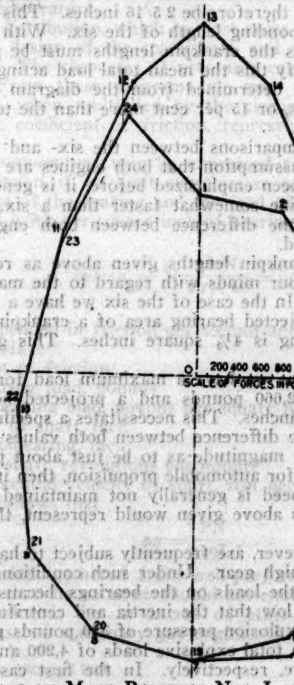


FIG. 7.—LOADS ON MAIN BEARINGS NOS. I AND VII OF CRANKSHAFT WITHOUT BALANCE WEIGHTS.

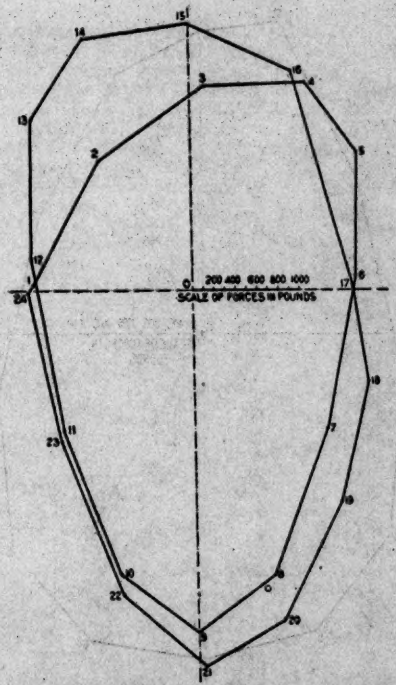


FIG. 8.—LOADS ON MAIN BEARINGS II, III, V AND VI OF CRANKSHAFT WITHOUT BALANCE WEIGHTS.

It is plainly evident from the scale of the diagrams that the balance weights act to reduce considerably the loads on the bearings, but, as already stated, magnitude is not the last word to be said about loads. Sudden fluctuations in the direction of the load are equally, if not more, detrimental to a bearing than the mere magnitude of the loads. Quick changes in the direction of a force acting on a mass are always accompanied by inertia effects. We must expect, therefore, that the journals carrying loads as shown in Figs. 10 to 12 will develop an undesired reciprocating or rocking action. Figs. 7 to 9, however, represent comparatively steady loads. The journals carrying these loads will creep around on the bearing surface, an action in principle like that involved when a planetary gear rolls within an internal gear. This facilitates lubrication, because the lubricant, while adhering to the metal surfaces, is continually wedged in between the two bounding surfaces. From this it is seen that beyond pounds of load per bearing there is, so far, nothing to say in favor of balance weights.

The length of the bearings necessary to carry the respective loads safely can be determined by the formula $pv = 17{,}000$ foot-pounds per second. With a main-bearing diameter of 2⅛ inches we obtain for 2,700 r.p.m. of

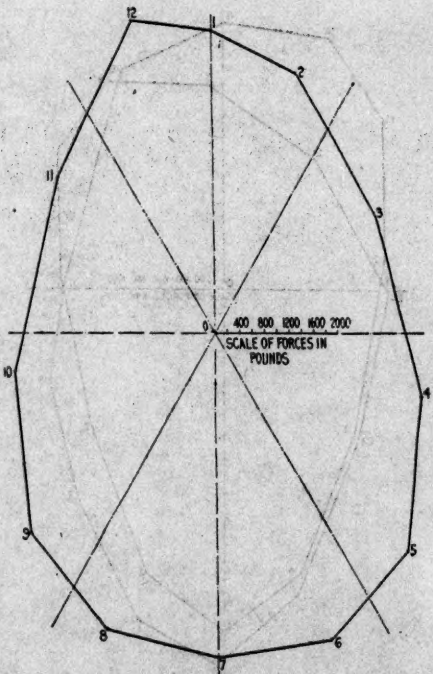


FIG. 9.—LOADS ON MAIN BEARINGS IV OF CRANKSHAFT WITHOUT BALANCE WEIGHTS.

the engine a circumferential velocity of $v = 25.02$ feet per second. The mean loads on the bearings, as obtained from the diagrams, are given in Table 2, together with the necessary bearing lengths, the values of pv and the specific bearing pressures.

In the case of bearings I and VII a liberal amount should be added to the length of the bearing to take care of the loads due to the timing gears and the flywheel, and at least ¼ inch should be added to every bearing to allow for fillets. But in spite of this we find that the bearing lengths for the balanced type are entirely too short to hold the requisite oil pressure. If they are lengthened so that they are in conformity with *this* practical consideration we will obtain bearings for a crank-shaft *with* balance weights about as long as those for a crankshaft *without* balance weights.

Apart from these analytical considerations it has now become an established fact that the smooth running, life and power of a well-designed engine with a seven-bearing shaft cannot be improved by the addition of balance weights. This is true in spite of the fact that any shaft with balance weights will perform much better on any balancing machine than will its prototype without balance weights. This paradoxical result is easily explained if we bear in mind that a shaft when running in a bal-

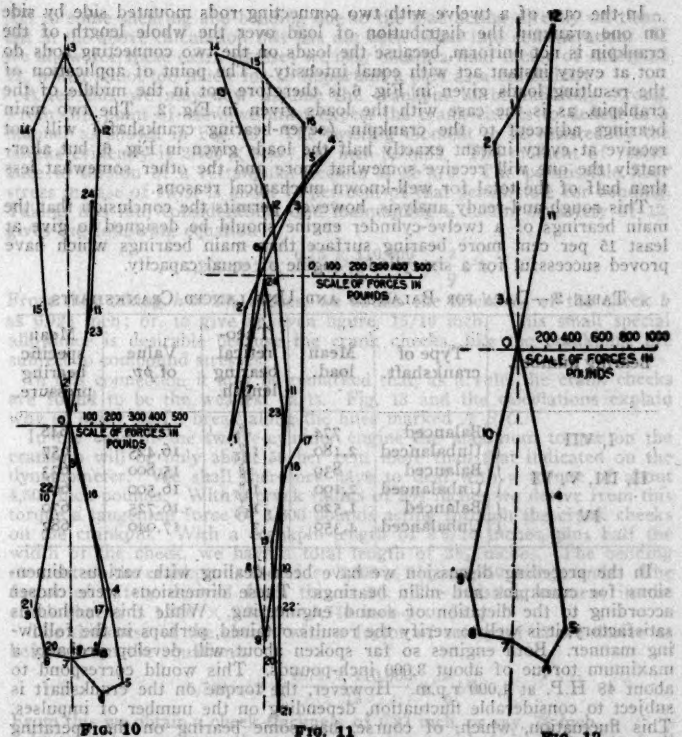

0 100 200 300 400 500
SCALE OF FORCES IN POUNDS
0 100 200 300 400 500
SCALE OF FORCES IN POUNDS
0 200 400 600 800 1000
SCALE OF FORCES IN POUNDS

FIG. 10 FIG. 11 FIG. 12

FIG. 10.—LOADS OF MAIN BEARINGS I AND VII. FIG. 11.—LOADS ON II, III, V AND VI OF A CRANKSHAFT WITH BALANCE WEIGHTS. FIG. 12.—LOADS ON IV.

ancing machine is not subject to the sudden impulses which are a necessary evil inherent to the reciprocating engine. It seems that the best method to make these impulses harmless is to smooth them out by means of the centrifugal forces, which, as we know, are available "free of charge."

It remains now to determine the loads on the main bearings for the twelve-cylinder engine in a manner similar to that which was adhered to in the case of the six. However, since there is not the slightest difference in the procedure, it is permissible to draw conclusions from former results. We must therefore reconcile ourselves with the fact that in all cases the mean pressure of a complete cycle will be about 15 per cent in excess of the corresponding pressures represented in Figs. 7 to 12. This conclusion is based on the results derived from the force diagrams shown in Figs. 2, 4 and 6. These diagrams show that the mean pressure on the crankpin of a twelve-cylinder engine is 15 per cent larger than that on the crankpin of a six.

In the case of a twelve with two connecting rods mounted side by side on one crankpin the distribution of load over the whole length of the crankpin is not uniform, because the loads on the two connecting rods do not at every instant act with equal intensity. The point of application of the resulting loads given in Fig. 6 is therefore not in the middle of the crankpin, as is the case with the loads given in Fig. 2. The two main bearings adjacent to the crankpin (seven-bearing crankshaft) will not receive at every instant exactly half the loads given in Fig. 6, but alternately the one will receive somewhat more and the other somewhat less than half of the total for well-known mechanical reasons.

This rough-and-ready analysis, however, permits the conclusion that the main bearings of a twelve-cylinder engine should be designed to give at least 15 per cent more bearing surface than main bearings which have proved successful for a six-cylinder engine of equal capacity.

TABLE 2.—DATA FOR BALANCED AND UNBALANCED CRANKSHAFTS.

Bearing number.	Type of crankshaft.	Mean load.	Theoretical bearing length.	Value of pv.	Mean specific bearing pressure.
I, VII	Balanced	774	$\frac{9}{16}$	16,200	648
	Unbalanced	2,180	$1\frac{9}{16}$	16,425	657
II, III, V, VI	Balanced	839	$\frac{4}{8}$	15,800	633
	Unbalanced	2,190	$1\frac{9}{16}$	16,500	660
IV	Balanced	1,520	$1\frac{1}{16}$	16,725	670
	Unbalanced	4,350	3	17,040	682

In the preceding discussion we have been dealing with various dimensions for crankpins and main bearings. These dimensions were chosen according to the dictation of sound engineering. While this method is satisfactory, it is well to verify the results obtained, perhaps in the following manner. Both engines so far spoken about will develop probably a maximum torque of about 3,000 inch-pounds. This would correspond to about 48 H.P. at 1,000 r.p.m. However, the torque on the crankshaft is subject to considerable fluctuation, depending on the number of impulses. This fluctuation, which, of course, has some bearing on the operating smoothness of an engine, was made the subject of many investigations in the early days of the six, and again at the advent of the eight and the twelve. Such investigations have revealed that the multi-cylinder engine, six, eight, twelve, etc., certainly deserves great credit for its smooth torque.

Owing to the small number of impulses, the torque on the crankpin of a six-cylinder engine may at intervals be two and a half times as large as the torque indicated on the dynamometer. In considering the strength of the crankshaft we must therefore deal with a torque of, say, 7,500 inch-pounds. With a crank radius of 3⅛ inches this torque results in a tangential force of 2,400 pounds, acting through the crank cheek on the crankpin as indicated in Fig. 13.

If we at first assume the crank cheeks as being infinitely strong, we can consider the crankpin as a cantilever. With a crankpin length of 2 inches, plus half the width of the cheek, the total length of the cantiliver will be about 2½ inches. The bending moment is, consequently, $2{,}400 \times 2\frac{1}{2} = 6{,}000$ inch-pounds. The moment of resistance to bending of a 2½-inch diameter cylinder with a 1-inch hole is 0.896 inch. With an elastic limit of 110,000 pounds per square inch (chrome-nickel steel) we obtain from these data 16.4 as the factor of safety. Crankpins are further sub-

ject to some shearing and bending due to the load acting directly on them, but in the above calculation this has not been taken into consideration. If we allow for these extra stresses we may obtain a final factor of safety of 12, which is not too much for so important a part as a crankshaft.

After this we may now assume the crankpin to be infinitely strong. From this point of view we must expect the crank cheeks to deflect in a manner as shown in Fig. 13. The twisting moment to be resisted by the cheeks evidently is $2{,}400 \times 3 = 7{,}200$ inch-pounds. The moment of resistance to torsion of a rectangular section is $2b^3h/9$. The permissible fiber stress in case of twist is about 75 per cent of that permissible for bending. Taking this into consideration, and maintaining a factor of safety of 12, gives the following equation:

$$7{,}200 = 2 \times \frac{0.75 \times 110{,}000}{12} \times \frac{2}{9} b^3h.$$

From this, taking $h = 2\frac{3}{4}$ inches, we obtain the thickness of the cheek b as 0.925 inch; or, to give an even figure, 15/16 inch. This small special allowance is desirable because the crank cheeks, like the crankpins, are subject to compound stresses.

In this connection it may be remarked that, as a rule, the crank cheeks are found to be the weakest parts. Fig. 13 and the calculations explain why so many shafts break along the lines marked *A B C*.

In the case of the twelve-cylinder engine the maximum torque on the crankpin will be only about 50 per cent more than that indicated on the dynamometer. We shall therefore have to deal with a torque of about 4,500 inch-pounds. With a crank radius of $2\frac{1}{2}$ inches we derive from this torque a tangential force of 1,800 pounds acting through the crank cheeks on the crankpin. With a crankpin length of 2 5/16 inches, plus half the width of the cheek, we have a total length of $2\frac{3}{4}$ inches. The bending moment on the crankpin is therefore $1{,}800 \times 2\frac{3}{4} = 4{,}950$ inch-pounds. The factor of safety derived from this is 20. The twisting moment in the crank cheeks is $1{,}800 \times 3\ 3/16 = 5{,}740$ inch-pounds.

The thickness of the cheeks with a factor of safety of 12 we derive as before from the equation:

$$5{,}740 = 2 \times \frac{0.75 \times 110{,}000}{12} \times \frac{2}{9} b^3h.$$

From this we obtain a cheek thickness of 0.83 inch, or say, $\frac{7}{8}$ inch.

In order to compare a three- and seven-bearing shaft it must first of all be observed that the distance between the centers of the two main bearings is about two and a half times as much in the former shaft as in the latter. The transverse deflection of a straight shaft increases as the cube of the distance between the supports. It is evident, therefore, that if other conditions are the same, a three-bearing shaft will deflect up to $(2\frac{1}{2})^3$; that is, nearly sixteen times as much as a seven-bearing shaft.

It is hardly necessary to point out that excessive deflections are certainly not conducive to the operating smoothness of an engine. Not only that, but even the durability of the bearings is impaired, because a deflected shaft will not permit uniform pressure distribution in the bearing. Any eccentricity of the loading relatively to the middle of the bearing will create a tendency to drive out the oil at one end. This is the reason why bearings that swivel so as to accommodate themselves to any inclination of the journal on account of bending of the shaft are found to be of great advantage. However, as such bearings cannot be applied very well to crankshafts, it is advisable to design crankshafts so that their deflections approach a minimum.

The deflection of a shaft varies inversely as the moment of inertia of its weakest section. In order, then, to hold a three-bearing shaft within the

same limit of transverse deflection as may be obtained with a seven-bearing shaft the crankpin diameters of both shafts must be proportioned in the relation of $\sqrt{10}$ to 1, which is as 2 to 1. This necessitates a crankpin diameter of 4¼ inches for a three-bearing shaft designed to conform to our premises. For so small an engine as here under consideration this diameter is, of course, prohibitive. At the same time, it is almost pro-

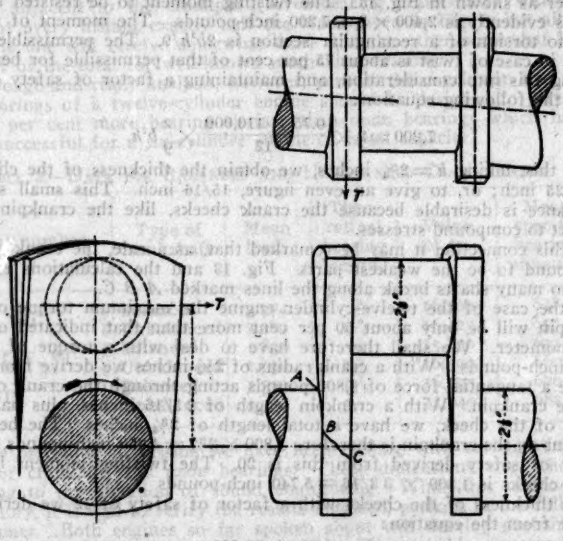


FIG. 13 PART OF CRANKSHAFT ILLUSTRATING DEFLECTION OF CRANK-CHEEKS

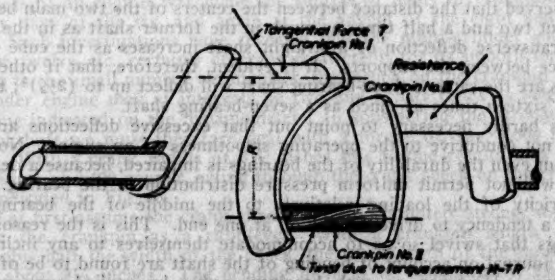


FIG. 14 PART OF CRANKSHAFT ILLUSTRATING TORQUE IN CRANKPIN IN CASE OF THREE- AND FOUR-BEARING SIX-THROW CRANKSHAFTS

hibitive to use a seven-bearing shaft for a twelve-cylinder automobile engine on account of the excessive length. The only alternative left for the designer of such engines is to sacrifice rigidity.

The torsional deflection of a crankshaft is directly proportional to the torque moment. In a seven-bearing crankshaft a torque moment of the magnitude $M = T \times r$ occurs in the main bearings only. This torque is determined by the tangential force T acting on the crankpin and by the crank radius r, which is one-half the stroke of the engine.

In a three-bearing crankshaft a similar but more intense torque occurs, not only in the main bearings but also in the crankpin, as illustrated in Fig. 14. The momentum of the torque in this case is determined by T and by the shortest distance between T and the center of the twisted crankpin, which is denoted by R. From Fig. 15 we find that $R = r \times r \sin 30°$

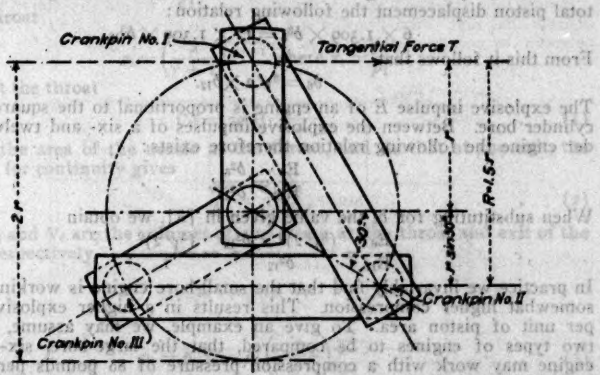


FIG 15.—PART OF CRANKSHAFT SHOWING MAXIMUM LEVERAGE IN CASE OF THREE- AND FOUR-BEARING SIX-THROW CRANKSHAFTS.

or $R = 1.5\ r$. Consequently, in this case, $M = T \times 1.5\ r$. This permits us to draw the conclusion that the torque moments or torsional deflections of a three- and a seven-bearing crankshaft stand to each other in the relation of 1.5 to 1. It must further be mentioned that the twist in a crankpin is a more serious matter than the twist in the main bearings. We must consider that the only way to reduce the vibrations set up by a twist in a crankpin is by an increase in its diameter. The deflections in the main bearings may, however, be minimized by vibration dampers.

The same line of reasoning which we have established in connection with the three-bearing shaft must be followed in investigating a four-bearing, six-throw shaft. The distance between two bearings is about 1.75 times as great as in the case of a seven-bearing shaft, and consequently the transverse deflection is about five times larger. With regard to the torque in the crankpins, the same figures as before established apply, except that in this case a smaller number of crankpins are subject to twist.

As excessive transverse deflections must be detrimental to the bearings as well as to the operating smoothness of the engine, it is quite reasonable to expect that balance weights will be an advantage for the latter two types of crankshafts, which are inherently weak. Seven-bearing crankshafts, when well designed, are inherently strong enough so that they cannot be improved through the addition of balance weights. We can predict, there-

fore, that the use of balance weights will be limited to shafts with a smaller number of bearings than would be desirable from the viewpoint of strength. Balance weights are like the flywheel, the symbol of some imperfection. If we compare shafts of equal strength we will further find that the weight of a three- or four-bearing, six-throw crankshaft that is properly balanced will by no means be less than that of a seven-bearing shaft.

The total piston displacements of different engines designed with a constant stroke-bore ratio are proportional to the cube of the cylinder bore. For instance, if the bore of an engine is b, and its stroke is $1.667b$, the piston displacement of this engine per cylinder is $(b^2\pi/4) \times 1.667b = 1.309b^3$.

The explosive impulses of different engines are proportional to the square of the cylinder bore, providing that the same compression is used throughout. Denoting the bore of a six-cylinder engine with b_6 and that of a twelve-cylinder engine with b_{12}, we find for two engines of equal total piston displacement the following relation:

$$6 \times 1.309 \times b_6^3 = 12 \times 1.309 \times b_{12}^3.$$

From this it follows that

$$b_6 = \sqrt[3]{2} \times b_{12}. \qquad (3)$$

The explosive impulse E of an engine is proportional to the square of the cylinder bore. Between the explosive impulses of a six- and twelve-cylinder engine the following relation therefore exists:

$$\frac{E_6}{E_{12}} = \frac{b_6^2}{b_{12}^2}.$$

When substituting for b_6 the value given in [3], we obtain

$$\frac{E_6}{E_{12}} = \frac{(\sqrt[3]{2})^2 \times b_{12}^2}{b_{12}^2} = \frac{(\sqrt[3]{2})^2}{1}.$$

In practice we invariably find that the small-bore engine is working under somewhat higher compression. This results in a higher explosive force per unit of piston area. To give an example, we may assume, for the two types of engines to be compared, that the large-bore, six-cylinder engine may work with a compression pressure of 85 pounds per square inch, and the small-bore twelve-cylinder engine with one of 90 pounds per square inch. Substituting this in the above given relation, we obtain

$$\frac{E_6}{E_{12}} = \frac{85}{90} \times \frac{(\sqrt[3]{2})^2}{1} = 1.5 \text{ to } 1.$$

From this we learn that for a certain standard of engineering the explosive impulse of a six-cylinder engine is about 1.5 times as large as that of a twelve-cylinder engine of equal total piston displacement.

In conclusion, a few words may be said about lubrication, although this can be considered a problem all by itself. To obtain absolutely satisfactory conditions, the oil delivery to the bearings must be in direct proportion to the work converted into heat by the bearings. We should lay out a series of diagrams, as previously discussed, for different engine speeds and for different power outputs. The values of pv obtained from the various diagrams would give us a clear picture of how the oil delivery should be regulated. Scientific oiling would eliminate scraper rings, and thus permit lighter pistons. It would also eliminate sooted spark plugs, carbonization, preignition, and would permit higher compression.—"Journal American Society of Mechanical Engineers."

THE DIVERGENCE OF STEAM NOZZLES.

By GERALD STONEY, F.R.S.

As in most of the text-books on steam turbines the theory of the divergence of steam nozzles is not fully given, the following method of calculation may be of interest.

In text books on steam turbines it is shown that in expanding adiabatically from a pressure p_1 to a pressure p_0 the velocity of the steam is

$$C_0^2 = 2g\,H_1 \frac{\gamma}{\gamma-1}\left(1 - z^{\frac{\gamma-1}{\gamma}}\right),$$

where

$$z = \frac{p_0}{p_1} \text{ and } H_1 = 144\,p_1\,V_1.$$

At the throat

$$z_t = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \text{ where } z_t = \frac{p_t}{p_1},$$

so that at the throat

$$C_t^2 = 2g\,H_1 \frac{\gamma}{\gamma+1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

If A_t is the area of the nozzle at the throat, and A_0 that at the exit, the equation for continuity gives

$$\frac{A_0}{A_t} = \frac{C_t\,V_0}{C_0\,V_t}, \quad . \quad . \quad . \quad . \quad (2)$$

where V_t and V_0 are the volumes of the steam at the throat and exit of the nozzle, respectively.

As

$$p_0\,V_0^\gamma = p_t\,V_t^\gamma$$

and as

$$p_t = p_1\,z_t = p_1\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} \quad . \quad . \quad . \quad . \quad (3)$$

we have

$$\frac{V_0}{V_t} = \left(\frac{p_t}{p_0}\right)^{\frac{1}{\gamma}} = \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}} z^{-\frac{1}{\gamma}}, \quad . \quad . \quad . \quad (4)$$

Putting equations (1), (2) and (4) into (5),

$$\frac{A_0^2}{A_t^2} = \frac{\gamma-1}{\gamma+1}\left(\frac{2}{\gamma+1}\right)^{\frac{2}{\gamma-1}} \frac{1}{z^{\frac{2}{\gamma}}\left(1 - z^{\frac{\gamma+1}{\gamma}}\right)}$$

$$= \frac{\gamma-1}{2}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}} \frac{1}{z^{\frac{2}{\gamma}} - z^{\frac{\gamma+1}{\gamma}}}, \quad . \quad . \quad . \quad (5)$$

an equation for the flare of the nozzle.

As Z is fractional, a more convenient form for calculation is obtained by writing $X = \frac{1}{z} = \frac{p_1}{p_0}$, X being the expansion by pressure; and then (5) becomes:

$$\frac{A_0^2}{A_t^2} = \frac{\gamma-1}{2}\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}} \frac{X^{\frac{\gamma+1}{\gamma}}}{X^{\frac{\gamma-1}{\gamma}} - 1}$$

As the ratio of the specific heats, or γ, may be considered as practically constant in these equations for the adiabatic expansion of steam, it will be seen that the flare depends only upon the amount of expansion by pressure in the nozzle; that is, that it is the same for a nozzle expanding from 200 pounds per square inch to 20 pounds as from 10 pounds to 1 pound. Since, however, the value of γ for saturated steam is less than for superheated steam, the flare is slightly less in the latter case.

For the critical expansion $Z = \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$ the value of A_0/A_1 is unity; and for the ordinary values of γ, which are from 1.135 for saturated steam to 1.3 for superheated steam, the relation between the flare and the ratio of expansion in the nozzle is approximately a straight line, except for large degrees of expansion.

It may be mentioned that in the common case, where the steam is at first superheated and them become wet as it expands, the average value of γ is between the value for superheated steam, namely, 1.3, and that for saturated steam, 1.135.

Fig. 1.

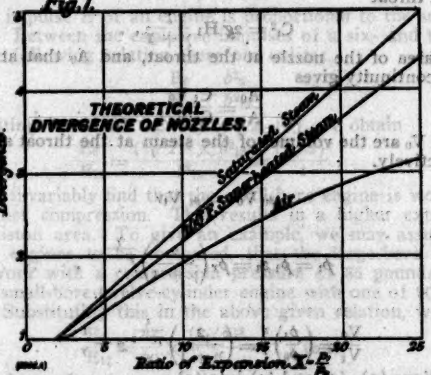


Fig. 2.

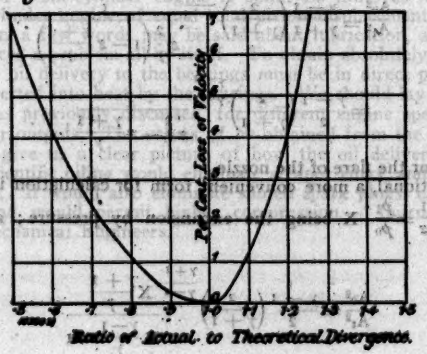


Fig. 1 shows the relation between the flare and the amount of expansion for saturated and superheated steam, and also for air. In this diagram "divergence" is to be understood as the ratio of area at any section of the nozzle to the throat area. In practice nozzles are generally given rather less flare, since it is much more important to have too little flare than too much flare.

That this is so, is well shown by the curves given by Mr. W. J. Goudie* from an analysis of data published by Dr. Steinmetz, a combination of these curves is given in Fig. 2. The effect of friction in a nozzle is generally to slightly increase the flare, but in practice this is negligible. Mr. H. M. Martin has pointed out to the author that

$$\frac{\lambda}{\lambda - 1} = \frac{1}{\eta} \frac{\gamma}{\gamma - 1}$$

where λ is, for frictional flow, the apparent value of the index in $pV^{\lambda} = \text{constant}$, and η is the efficiency. For many purposes this formula is more conveniently written as

$$1 - \frac{1}{\lambda} = \eta \left(1 - \frac{1}{\gamma} \right).$$

As in practice the efficiency is always above 90 per cent, it is easily seen that the losses in the nozzle have little effect on the flare.—"Engineering."

FABRICATED SHIPS.†

METHODS OF CONSTRUCTION DEVELOPED BY CHESTER SHIPBUILDING COMPANY.

By R. H. M. Robinson.

Recently the term "fabricated ships" has become one of public interest. A fabricated ship may be defined, briefly, as a ship on which the work of punching and shaping the plates and shapes and, to some extent, assembling and riveting, is done in a fabricating shop, ordinarily employed for bridge or tank work, as distinguished from the usual practice of doing it in a shipyard punch shop.

So far as the writer knows, the fabricated ship is a product of American progressiveness. The principal credit for carrying out the fabricating idea is due to Mr. C. P. M. Jack, now consulting engineer, and to Mr. Max Willemstyn, engineering manager of the Chester Shipbuilding Company and of the Merchant Shipbuilding Corporation, and it would be improper to neglect to state that the ideas of Messrs. Jack and Willemstyn could probably not have been carried out except for the interest and assistance given them by Mr. Jas. A. Farrell, president of the United States Steel Corporation, and by the officials of the American Bridge Company.

The construction of fabricated ships makes it possible to have the steel work done by those who are specialists in the line of fabricating steel with all of the special tools and labor-saving devices at their disposal. It relieves the mind of the shipyard operator from the multitude of detail that goes with such work, reducing his problems to those of engineering, erection, riveting and assembling by the workmen, and to the installation of other units of outfit and equipment.

*"Steam Turbines," by W. J. Goudie, p. 133.

† Paper read before the Society of Naval Architects and Marine Engineers, New York.

SHIP SHAPES AND PLATES TO BE USED.

The type of ship which the Chester Shipbuilding Company has built, and which it and the Merchant Shipbuilding Corporation are now building, provides for the use in all places of conventional ship shapes and ship plates, subject to the rules, inspection and survey of the registration society, both as to quality and material, dimensions, riveting and other matters. These vessels are cargo carriers of about 8,800 to 9,000 tons deadweight capacity with poop, topgallant forecastle and bridge, the latter being about 124 feet long. The decks, with the exception of the poop and topgallant forecastle, are without sheer at the center line. Complete double bottom for the carrying of fuel oil or water ballast is fitted. The machinery is located amidships and consists of high and low-pressure turbines, driving a single screw through reduction gears.

In the emergency which has recently arisen there have been proposed ships constructed from bridge shapes and universal mill plates, with regard to which the writer has no personal detailed knowledge. It would appear to him that there might arise certain difficulties in using this character of material in the construction of ships, but he would prefer to leave to those who are personally cognizant of the matter any questions in regard to this.

The manufactured ship is a commercial possibility only if duplicate ships are built and, therefore, this policy was adhered to as far as possible from the beginning in contracting for the ships at Chester, there being but four different classes. These classes are as follows:

TYPES OF SHIPS.

1. Tank ship of about 9,000 tons deadweight capacity, fitted with cylindrical tanks on Jack's system. The vessel is built with topgallant forecastle, poop and short bridge. The main deck at centerline and top of expansion trunk are without sheer. Cylindrical tanks extend from tank top to under side of main deck, and a continuous trunk extending between poop and topgallant forecastle takes care of expansion and provides a working deck between the bridge, poop and forecastle decks. Machinery is located aft and consists of a high and low-pressure turbine driving a single screw through reduction gears.

2. Two-decked freighter of about 8,800 to 9,000 tons deadweight capacity, having poop, topgallant forecastle and bridge, the latter being about 124 feet in length. This vessel is built with sheer. Complete double bottom for the carrying of fuel oil or water ballast is fitted. Machinery is located amidships and consists of high and low-pressure turbines driving a single screw through reduction gears.

3. Shelter-deck freighter of about 9,300 tons dead-weight capacity, having poop, topgallant forecastle and bridge. Complete double bottom for the carrying of fuel oil or water ballast is fitted. Machinery is located amidships and consists of high and low-pressure turbines driving a single screw through reduction gears.

4. Two-decked freighter of about 8,800 to 9,000 tons deadweight capacity, having poop, topgallant forecastle and bridge. The decks, with the exception of the poop and forecastle decks, are without sheer at the centerline. Complete double bottom for the carrying of fuel oil or water ballast is fitted. Machinery is located amidships and consists of high and low-pressure turbines, driving a single screw through reduction gears.

Except as to details which may be modified to provide for ease of fabrication, the fabricated ship does not differ essentially in structure from any ship of a similar class. One of the prime questions involved is that of multiple punching.

MULTIPLE PUNCHING.

The greatest difficulty with the arrangement for multiple punching lies in the fact that the minimum distance from center to center of punch dies in a multiple punch is regularly 2 inches, which distance can occasionally be reduced to 1⅞ inches.

The term "multiple punch" implies that the spacing of rivets is to be laid out so that the spacings of rivets in different successive rows are in multiples of each other or in multiples of 2 inches. Take for example a butt spacing of four diameters of ⅞-inch rivets equal to 3½ inches. The frame riveting in this case could be six diameters or 5¼ inches, which is no multiple of 3½ inches, and neither is 3½ inches a multiple of 2 inches.

A compromise is made in this case by making an average butt-spacing of 3½ inches, which can be done by spacing in groups of three, viz: 3¼ inches, 4 inches, 3¼ inches. The punches for the butt and frame spacing would be spaced in the machine as follows: 3¼ inches, 2 inches, 2 inches, 3¼ inches. The first, second, fourth and fifth punches would be used for the butt-spacing. The first, third and fifth for the frame spacing allowing for a 5¼-inch spacing.

Where the butt-spacing is 3½ diameters and the frame spacing 7 diameters, such as on smaller size ships, there is not much trouble, but the multiple punch is hardest to apply on the 9,000-ton, clear-hold, single-deck ships, with their 5½ and 6 diameters frame riveting. Again, on the larger ships with 4-inch butt-spacing it is easier to be according to rule, and it should be here noted that Lloyd's Register of Shipping has been very progressive and broad-minded in allowing the use of the average spacing instead of the strict and absolute adherence to stated rules.

The first thing to do on a ship designed for multiple punch is to arrange the midship section in such a way that all longitudinal rows of holes fall in line with the principles just explained; second, to arrange the plate width so that the greatest number of equal width plates are obtained; third, to see that, if possible, each plate is symmetrical about the long centerline of the plate, so as to avoid any possible countersinking errors. No templates are required for setting the multiple punch, which is entirely done from the typical rivet-spacing drawing.

In staggered riveting, the minimum longitudinal spacing between the rivets of one row and those of the other is 1 inch, but 1¼ inches is preferable. This space is required for the stops and pawls, which stop the table during the downward stroke of the punch-head. It is, of course, possible to reduce the minimum space to small fractions, but then the human element of error is introduced, as a man has to insert a liner between stop and pawl, equal in thickness to the change in spacing wanted. Otherwise the machine is electrically controlled, except the throwing in of the clutch for punching.

A "pole," being a batten about 3 inches wide by ½ inch thick and of such length as is required, is used as template for the longitudinal spacing. Steel stops are inserted in the holes of the batten corresponding with rivet spacing, or, on the newest machines, steel stops are set up in a steel rack to suit the pole. Different color of light is shown where butts, edge-seams, frames, etc., occur, and the man on the levers is acting solely according to these colors.

The multiple punches have, as a rule, only three or four set-ups; that is to say, one set-up for butts, one for edge-laps, one for frames, one for, say, intercostal angles in conjunction with the edge-laps. In this case the watertight bulkheads could not be punched if the holes would not entirely coincide with one of the four set-ups. It is possible to omit a few holes in some of the set-ups, but this is done only by hand. The man on the punch can release any number of punches temporarily by disconnecting the gag-operating gear for these punch dies.

The Chester Shipbuilding Company, Ltd., has, wherever possible, carried out the idea of duplication and standardization, but very often has had to abandon it in the detail. The hull, engines, boilers, and, in fact, all large items for a group of ships, have been duplicated.

The dimensions for all but two ships contracted for by the Chester Shipbuilding Company are: Length, 401 feet; molded breadth, 54 feet; molded depth, 32 feet 9 inches. Some have had sheer, but, in the latest type, sheer is dispensed with and a raking forecastle and poop substituted, giving the effect of sheer, but simplifying the work of manufacture.

The results of the fabrication in the Bridge Company's mills as outlined above is beyond criticism. Work is fair, rivet holes require no reaming, and the resulting fit of joints and watertightness is excellent. Constructing a ship becomes to a great extent a problem of routing, handling and erection of material and of riveting.

The Chester Shipbuilding Company, Ltd., and the Merchant Shipbuilding Corporation are now building some sixty-eight ships on the manufacturing principles outlined above. It is expected to embody even more than has been possible in the past the manufacturing idea, thus making it possible, in this day of labor shortage, to do a maximum of work with a minimum of skilled shipyard labor of the conventional kind.—" Marine Engineering."

CARGO-SHIP LINES OF SIMPLE FORM.

PAPER READ BEFORE NAVAL ARCHITECTS' SOCIETY DESCRIBING RESULTS OF TESTS OF MODELS OF SIMPLIFIED FORM AT WASHINGTON.

By Naval Constructor William McEntee, U. S. N.

The recent wide discussion and general interest in possibilities of standard cargo ships to be built in large numbers from standard plans has naturally brought into prominence the matter of simplifying the form of ships.

The tendency in ship construction for a long period has been in the direction of simplification. In general, the simplification of form requires the use of plain surfaces at right angles instead of curved surfaces wherever possible. Where curvature is unavoidable, the simplest of curves, the circle, should be used. All curvature should be in one direction only; that is, surfaces of double curvature should be avoided so far as possible.

These considerations lead, in the forming of a ship, to the adoption of a midship section formed by two vertical lines and a horizontal line at the bottom, the corners or bilges being rounded off by circles of suitable radius. The sides and bottom of the ship will be flat surfaces carrying the full midship section well forward and aft, making what is known as the parallel middle body extend as far as possible without excessive increase in resistance. Forward, where it is necessary to fine the lines to form an entrance, it can be done by gradually decreasing the width of the ship, keeping the bottom flat and the sides vertical, the bilge being formed by a quarter-circle of constant radius equal to that used for the middle part of the vessel.

Aft, the possibilities of simplification are limited by the necessity for providing space for the propellers. Two different methods of simplifying the after body are shown in Model 1,997 and Model 2,056, Figs. 1 and 2. The general result in these forms is to make the waterline somewhat wider and place the displacement higher up. This arrangement seems favorable from both the resistance and the propulsion standpoints, but may have disadvantages as regard seagoing qualities.

TABLE I—DIMENSIONS

Model Number.	Type.	Length. Feet.	Beam. Feet.	Draft Feet.	Displacement. Tons.	Speed. Knots.	E. H. P.	Longitudinal Coefficient.	Midship Section Coefficient.	Parallel Middle Body. Percent.
1978	Oil fuel barge, conventional form	160	25	9.25	830	8	34.4	.62	.98	38
1997	Oil fuel barge, simplified lines	160	25	9.25	830	8	31.6	.62	.98	38
2025	400-foot freighter, conventional form	400	57.3	24	13,157	10.5	1999	.738	.987	24
2049	400-foot freighter, simplified lines	400	57.3	24	13,157	10.5	1158	.738	.987	24
Series*	400-foot cargo vessel, conventional form	400	72.6	24.77	20,000	14	3460	.74	.98	38
2045	400-foot cargo vessel, simplified lines	400	72.5	24.375	20,000	14	3280	.74	.988	38
1121	U. S. Collier Neptune	450	65	27.5	19,340	14	3460	.738	.983	38

* Estimated from Experimental Model Basin's Parallel Middle Body Series (No. 24).

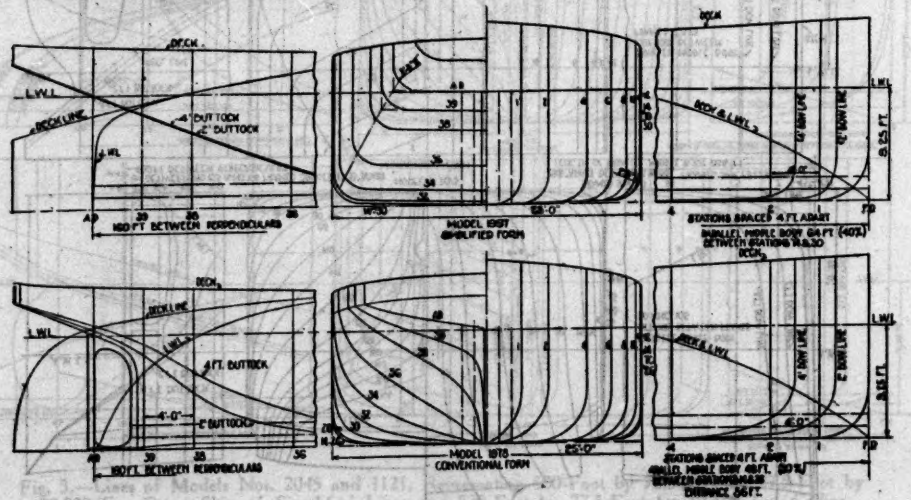


Fig. 1.—Lines of Models Nos. 1978 and 1997, Representing 160-Foot by 25-Foot by 9.25-Foot by 830-Ton Fuel Oil Barges of Conventional Form and of Simplified Lines

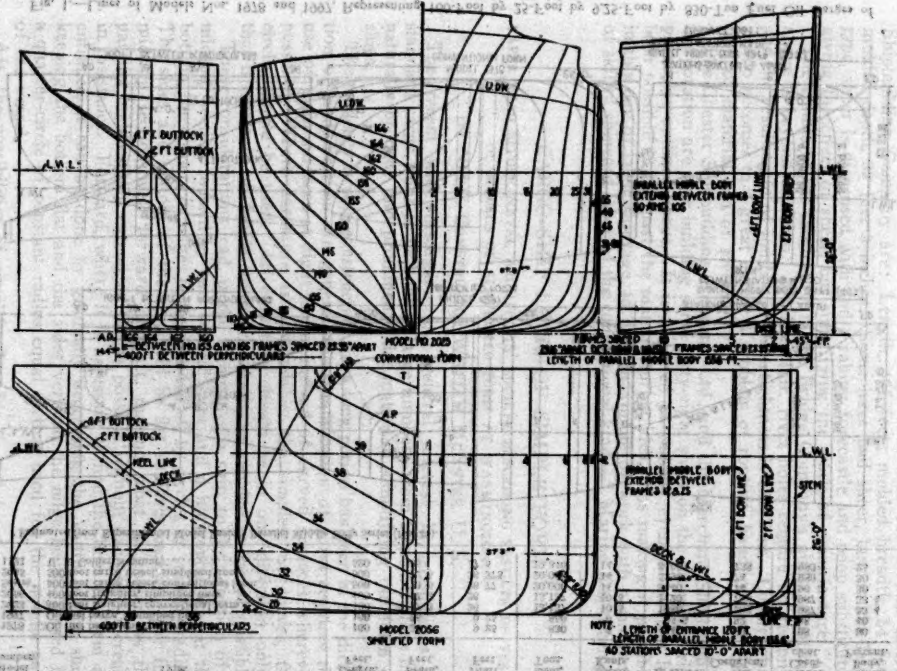


Fig. 2.—Lines of Models Nos. 2023 and 2056, Representing 400-Foot by 57.3-Foot by 26-Foot by 13,137-Ton Cargo Ships of Conventional Form and of Simplified Lines

Fig. 2.—Lines of Models Nos. 2023 and 2056, Representing 400-Foot by 57.3-Foot by 26-Foot by 13,137-Ton Cargo Ships of Conventional Form and of Simplified Lines

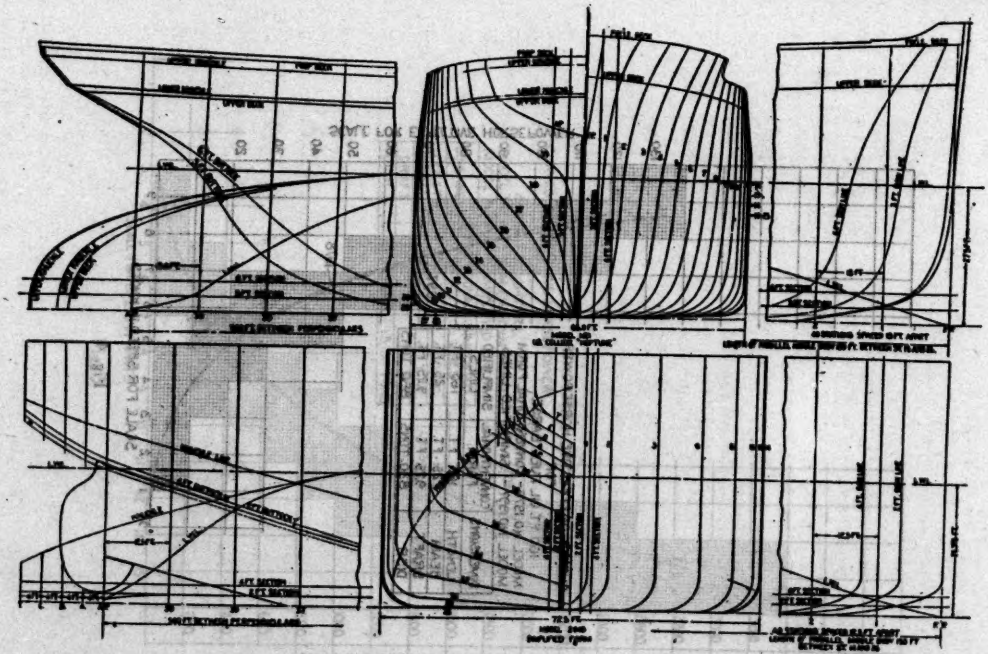


Fig. 3.—Lines of Models Nos. 2045 and 1121, Representing 500-Foot by 72.5-Foot by 27.5-Foot by 20,000-Ton Cargo Ship of Simplified Lines and 520-Foot by 27.5-Foot by 19,340-Ton Collier

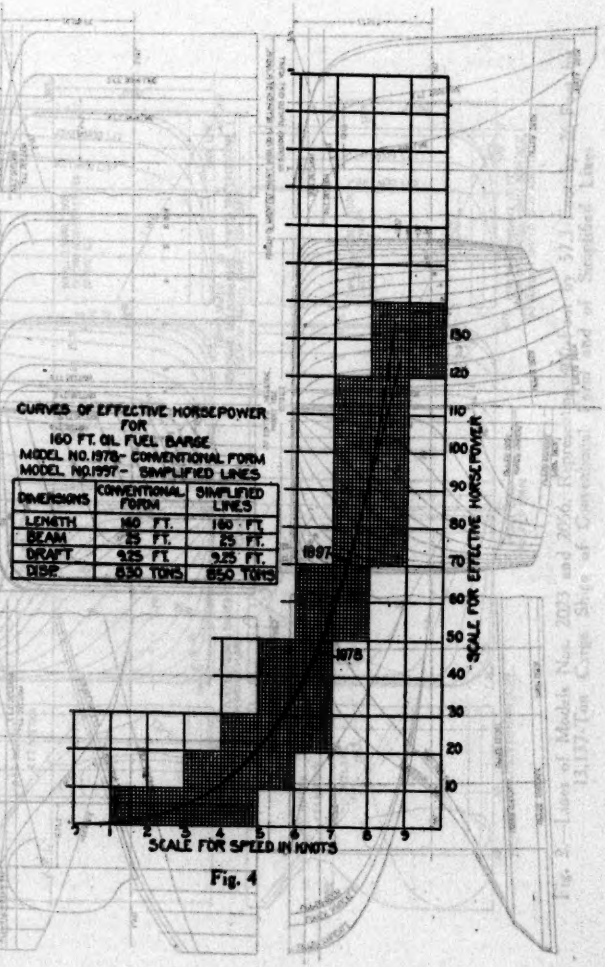

CURVES OF EFFECTIVE HORSEPOWER
FOR
160 FT. OIL FUEL BARGE
MODEL NO. 1978 - CONVENTIONAL FORM
MODEL NO. 1997 - SIMPLIFIED LINES
DIMENSIONS | CONVENTIONAL FORM | SIMPLIFIED LINES
LENGTH | 160 FT. | 160 FT.
BEAM | 25 FT. | 25 FT.
DRAFT | 9.25 FT. | 9.25 FT.
DISP. | 6.30 TONS | 6.50 TONS
1997
1978
SCALE FOR EFFECTIVE HORSEPOWER
10
20
30
40
50
60
70
80
90
100
110
120
130
SCALE FOR SPEED IN KNOTS
1
2
3
4
5
6
7
8
9

Fig. 4

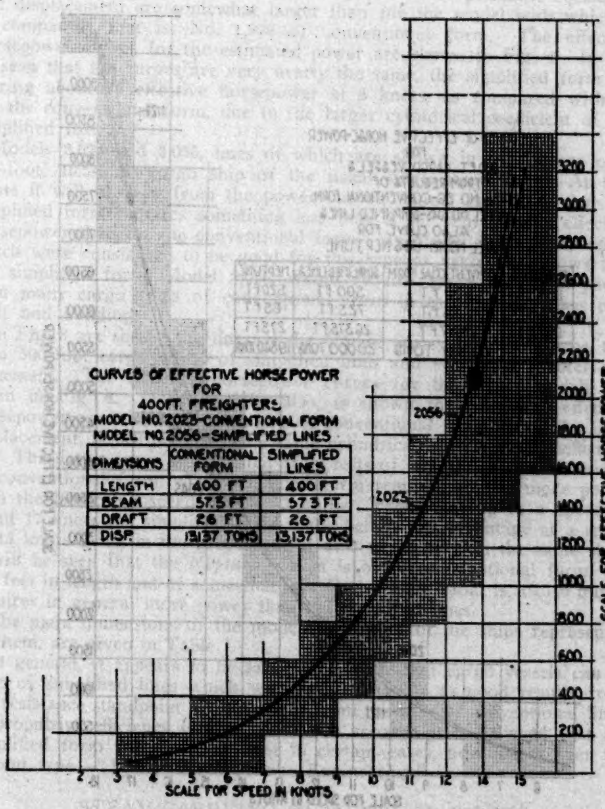

CURVES OF EFFECTIVE HORSEPOWER
FOR
400FT. FREIGHTERS.
MODEL NO. 2023-CONVENTIONAL FORM
MODEL NO. 2056-SIMPLIFIED LINES
DIMENSIONS | CONVENTIONAL FORM | SIMPLIFIED LINES
LENGTH | 400 FT | 400 FT
BEAM | 57.5 FT | 57.3 FT.
DRAFT | 26 FT. | 26 FT
DISP. | 13137 TONS | 13,137 TONS
2056
2023
SCALE FOR EFFECTIVE HORSEPOWER
SCALE FOR SPEED IN KNOTS

Fig. 5

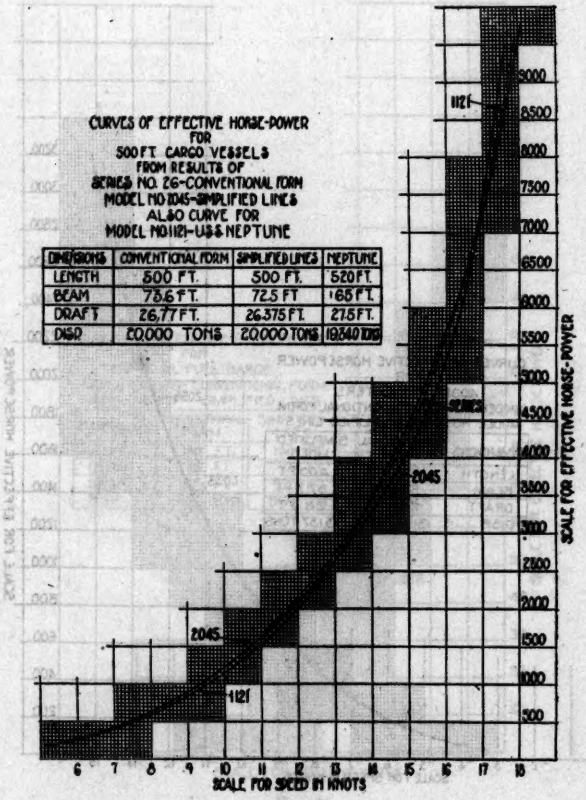

CURVES OF EFFECTIVE HORSE-POWER
FOR
500 FT CARGO VESSELS
FROM RESULTS OF
SERIES NO. 26-CONVENTIONAL FORM
MODEL NO 1045-SIMPLIFIED LINES
ALSO CURVE FOR
MODEL NO 1121-U.S.S. NEPTUNE
DIMENSIONS | CONVENTIONAL FORM | SIMPLIFIED LINES | NEPTUNE
LENGTH | 500 FT. | 500 FT. | 520 FT.
BEAM | 78.6 FT. | 72.5 FT | 68 FT.
DRAFT | 26.77 FT. | 26.375 FT. | 27.5 FT.
DISP | 20,000 TONS | 20,000 TONS | 19,340 TONS
1121
SERIES
2045
2045
1121
9000
8500
8000
7500
7000
6500
6000
5500
5000
4500
4000
3500
3000
2500
2000
1500
1000
500
SCALE FOR EFFECTIVE HORSE-POWER
6 7 8 9 10 11 12 13 14 15 16 17 18
SCALE FOR SPEED IN KNOTS

Fig. 6

In the present investigation an attempt has been made to show how the power required for propulsion is affected by simplification of the lines of three different types of cargo vessel.

Models 1,978 and 1,997, lines of which are shown on Fig. 1, are models of a 160-foot oil-fuel barge of 830 and 850 tons, respectively. Through a slight error in making Model 1,997, the cylindrical or prismatic coefficient and displacement are somewhat larger than for the model with which it is compared; that is, No. 1,978 of conventional form. The effective horsepower curves for the estimated power are shown in Fig. 4. It will be seen that the curves are very nearly the same, the simplified form requiring about 35 effective horsepower at 6 knots, as compared with 34 for the conventional form, due to the larger cylindrical coefficient of the simplified form.

Models 2,023 and 2,056, lines of which are shown in Fig. 2, are of a 400-foot, 10.5-knot cargo ship of the usual single-screw type. At 10.5 knots it will be seen from the power curves, shown in Fig. 5, that the simplified form requires something less than 2 per cent greater effective horsepower than for the conventional form. As Model 2,023 gave results which were considered to be good for this type of vessel, it follows that the simplified form, Model 2,056, would require appreciably less power than many cargo ships of conventional form in this general class now built and building.

In Fig. 3 are shown the lines of the United States collier *Neptune*, and of a 500-foot cargo vessel of simplified lines and somewhat greater displacement. The effective horsepower curves for these two models are given in Fig. 6. On this plate, also, is shown the estimated effective horsepower curve for a 500-foot ship of conventional form having the same displacement, 20,000 tons, and the same cylindrical or prismatic coefficient, .74. This latter curve is estimated from tests of one of a series of models of conventional form. The simplified form requires a little more power than the latter for speeds up to 13 knots. From that up to a speed of about 17 knots the simplified form is the better, the advantage at a speed of 15 knots being in the neighborhood of 7 per cent. On the same sheet it will be seen that the *Neptune*, which is of the conventional form, but 520 feet in length and of somewhat less displacement—that is, 19,340 tons—requires in general more power than the simplified lines.

The main dimensions of the models, or rather of the ships represented by them, are given in Table I.

In general, it appears to be safe to conclude that cargo vessels can be built of simplified lines which will give practically as good results from the resistance standpoint as those built from the present conventional lines. If propulsive efficiency is also taken into account, it is believed that the simplified form will have, at least in certain cases, advantages over the present type.—"International Marine Engineering."

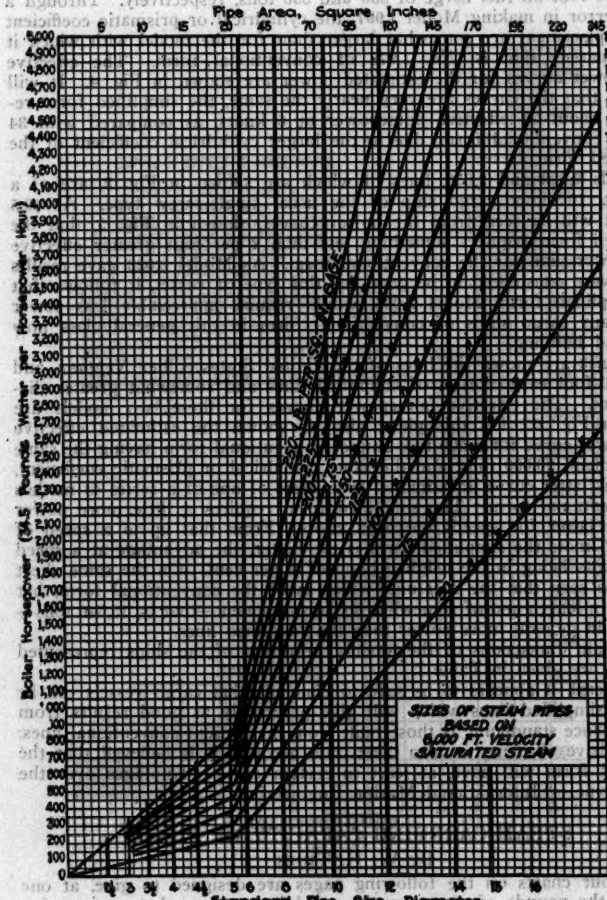
STEAM-CARRYING CAPACITY OF PIPES.

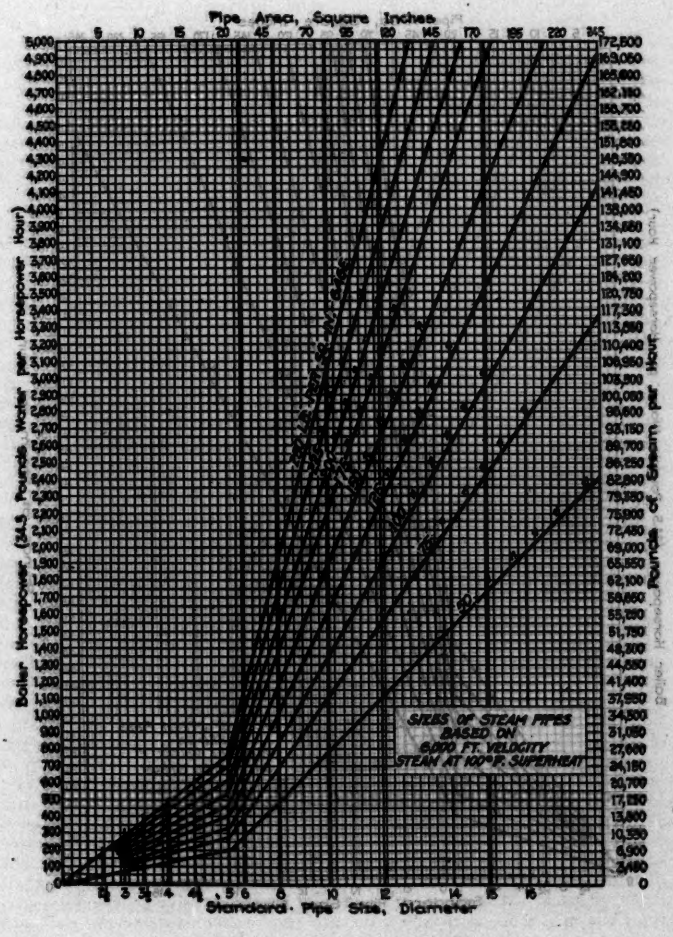
By W. H. Thies.

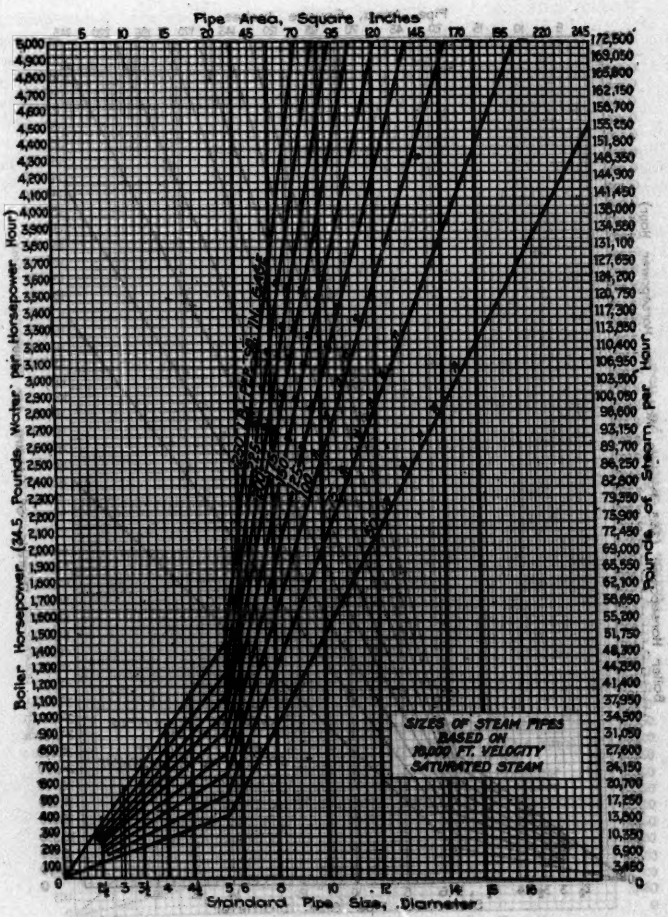
The four charts on the following pages are designed to give, at one reading, the pounds of steam or the rated horsepower that a pipe of a given size will supply under the conditions of pressure, heat and velocity given. The two velocities and temperature conditions used offer sufficient range from which to determine the pipe size for almost any other condition as near as the commercial pipe sizes will permit.

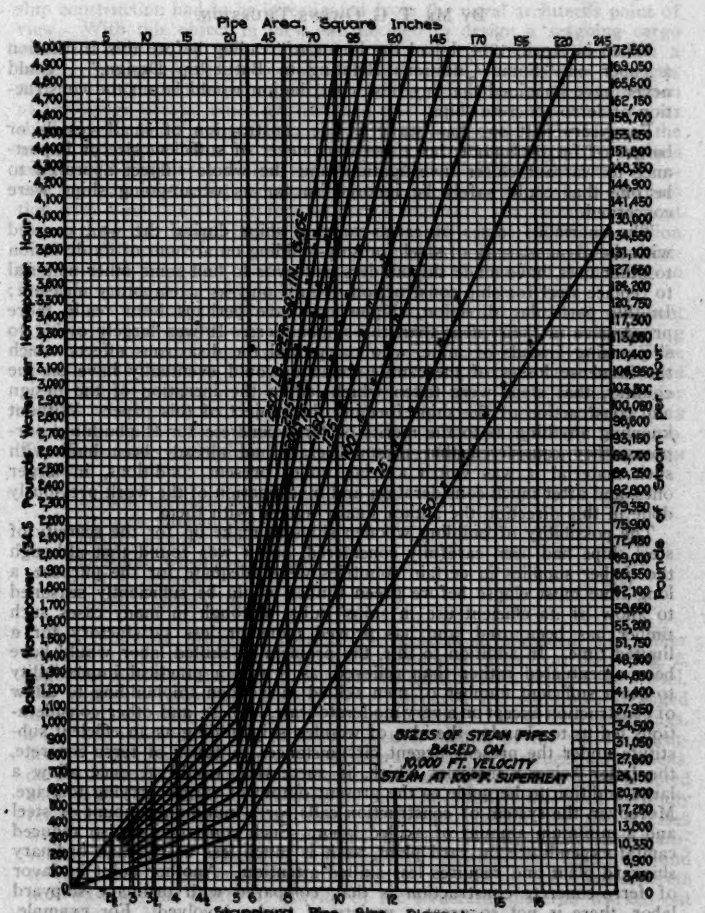
It is well to remember that pipe sizes are only nominal at best and that 14-inch and over is "outside diameter."

The charts have been found extremely useful. They are entirely original and have never before been published.—"Power."


Pipe Area, Square Inches
Boiler Horsepower (34.5 Pounds Water per Horsepower Hour)
Pounds of Steam per Hour
Standard Pipe Size, Diameter
SIZES OF STEAM PIPES BASED ON 6,000 FT. VELOCITY SATURATED STEAM


Pipe Areas, Square Inches
Boiler Horsepower (34.5 Pounds Water per Horsepower Hour)
Pounds of Steam per Hour
Standard Pipe Size, Diameter
SIZES OF STEAM PIPES
BASED ON
6,000 FT. VELOCITY
STEAM AT 100° F. SUPERHEAT


Pipe Area, Square Inches
5 10 15 20 45 70 95 120 145 170 195 220 245
Boiler Horsepower (34.5 Pounds Water per Horsepower Hour)
Pounds Steam per Hour
Standard Pipe Size, Diameter
SIZES OF STEAM PIPES
BASED ON
10,000 FT. VELOCITY
SATURATED STEAM


Pipe Area, Square Inches
5 10 15 20 45 70 95 120 145 170 195 220 245
Boiler Horsepower (34.5 Pounds Water per Horsepower Hour)
Pounds of Steam per Hour
Standard Pipe Size, Diameter
SIZES OF STEAM PIPES
BASED ON
10,000 FT. VELOCITY
STEAM AT 100° F. SUPERHEAT

REINFORCED CONCRETE SEAGOING CARGO STEAMERS

ON THE DESIGN AND CONSTRUCTION OF SELF-PROPELLED REINFORCED CONCRETE SEAGOING CARGO STEAMERS, NOW BUILDING IN GREAT BRITAIN.*

By Mr. T. G. Owens Thurston.

When in April, 1914, I had the honor of reading before this Institution a paper on "Some Questions Relating to Battleship Design," I could never have imagined that my next contribution would be on the construction of ferro-concrete ships.

For years past one has heard of the construction of small vessels or barges of ferro-concrete, but these were never of sufficient size or importance to warrant closer investigation, and the whole subject appeared to be one that could safely be ignored so far as ocean-going ships were concerned.

The enormous losses of cargo-carrying ships during the war, coupled with the great scarcity of steel for ship-building on account of its diversion to other uses, have made shipbuilders endeavor to find some other material to replace steel for ship construction, even if only as a temporary measure; In this particular direction, shipowners have perhaps been really more progressive in their ideas than the shipbuilders. It was largely owing to shipowning friends of mine, who recognized the disastrous effects which the immense losses of mercantile tonnage would eventually have on the country, that steps were taken which led to the inception of the design and construction of the ships forming the subject of this paper. Without knowing anything as to the merits of ferro-concrete for shipbuilding purposes, they considered that if small craft had already been built with such material it was worth a trial, under present conditions, in larger ones, and after much consideration and investigation of the work previously done in this direction the author finally agreed with them.

The possibility of building wooden ships to make up for the scarcity of steel ships had been carefully considered, but it was found that although the timber supplies in certain countries were adequate for the purpose, a long time must elapse before green timber would be sufficiently seasoned to allow of its efficient use in ships' hulls, although in many cases such timber has been used, and has proved more or less satisfactory for a limited time. In addition to this, the problem of skilled labor would have been accentuated rather than relieved, as it was a practical impossibility to get a sufficient number of men skilled in wooden shipbuilding to allow of construction except on a very moderate scale. These and other considerations led us to abandon the idea of wooden shipbuilding as an efficient substitute under the present urgent circumstances in favor of ferro-concrete, the latter being a material which, if successful, would at once allow a large addition to be made to the present output of cargo-carrying tonnage. Moreover, the system of construction calls for a minimum amount of steel and a minimum amount of skilled labor. Such construction also reduced capital expenditure on yard plant, as it is much less costly than ordinary shipyard plant and requires less skilled attention. Another point in favor of ferro-concrete construction is that, compared with ordinary shipyard labor, there is not so great a variety of trades involved. For example, the same men who, at a later date, are employed on casting the hull, are in the initial stages utilized in casting and laying concrete blocks and in making ferro-concrete piles for the building berths and other necessary

* Paper read at the Spring meeting of the fifty-ninth session of the Institution of Naval Architects, March 22, 1918.

preliminary work, for which this material can be used extensively instead of timber or steel, both of which require skilled labor of different trades.

Although the suitability of ferro-concrete under present abnormal conditions and on economical considerations was apparent, its acceptability for ship construction had to be investigated from the naval architect's point of view. With this object in view, we decided to design a seagoing cargo vessel of reasonably large dimensions to assure ourselves that such a vessel, constructed of ferro-concrete, would satisfy conditions of strength, seaworthiness, deadweight capacity, etc. The difficulties in the preparation of the design proved much greater than was anticipated, for whilst we, as shipbuilders, could prepare and supply the particulars and drawings of the vessel and approximate to the maximum stresses coming on the various members, we had no practical experience of the construction of the ferro-concrete part of the hull, so that it became necessary to work in conjunction with a reliable firm of ferro-concrete engineers, who would carry out this part of the design, basing it on work actually done. This association of shipbuilders and ferro-concrete engineers has proved mutually satisfactory and advantageous, and each party found that it had something to learn from the other with respect to shipbuilding of this type.

It was eventually decided that a self-propelled cargo vessel of about 1,150 tons deadweight was as large as we were justified in commencing as a first venture. It was agreed that the vessel should comply with all the requirements obtaining for steel vessels, and the general scheme was based upon that of a similar ship constructed of steel. The dimensions, for reasons which occurred in preparing the design, differed from those of a steel ship of the same deadweight carrying capacity, especially in length. The dimensions and other particulars finally decided upon were as given below:

Length between perpendiculars, feet	205
Breadth molded, feet	33
Depth, molded, feet and inches	19–6
Draught, when loaded, feet and inches	15–6
I.H.P. (about)	400
Speed (about), knots	7¾

The following drawings indicate the general arrangement and other features of the vessel Fig. 1, page 424, general arrangement plan; Fig. 2, page 425, midship section; Fig. 3, sheer draught or lines plan of the vessel. The various arrangement drawings were prepared and scantlings arranged in accordance with Lloyd's Rules for steel ships. Using these scantlings, a list was made up of the calculated section moduli of the various members. Upon these particulars the engineers prepared constructional sections, on the Hennebique principle of "equivalent strength," which were adopted as a basis for calculation of weights, etc. The design was then carefully reconsidered and calculations made as far as possible for the stresses which the vessel would be subjected to, both longitudinally and transversely, under all reasonable conditions of construction, launching and service. One of the first things observed in the case of the ferro-concrete hull was that the center of gravity of the structural material worked out much lower than in the case of a steel hull, the result being a greater metacentric height than was desirable. To minimize this, the original beam of 34 feet was reduced to 33 feet, and in order to obtain the same hold capacity the depth was increased from the original 17 feet 6 inches to 19 feet 6 inches, fortunately, from the point of view of longitudinal strength, a modification in the right direction.

Fig.1.

LENGTH B.P. 205'-0" × 32'-0" × 19'-6" MLD. DEPTH

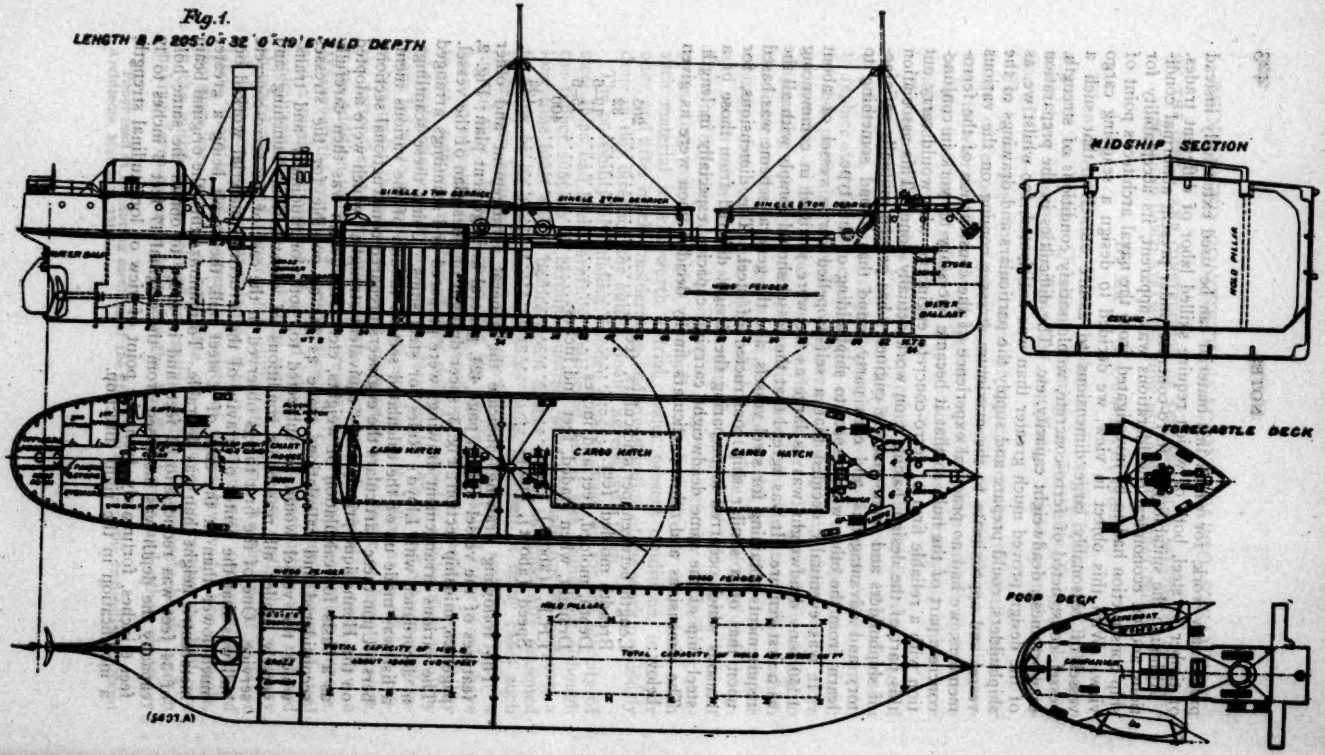
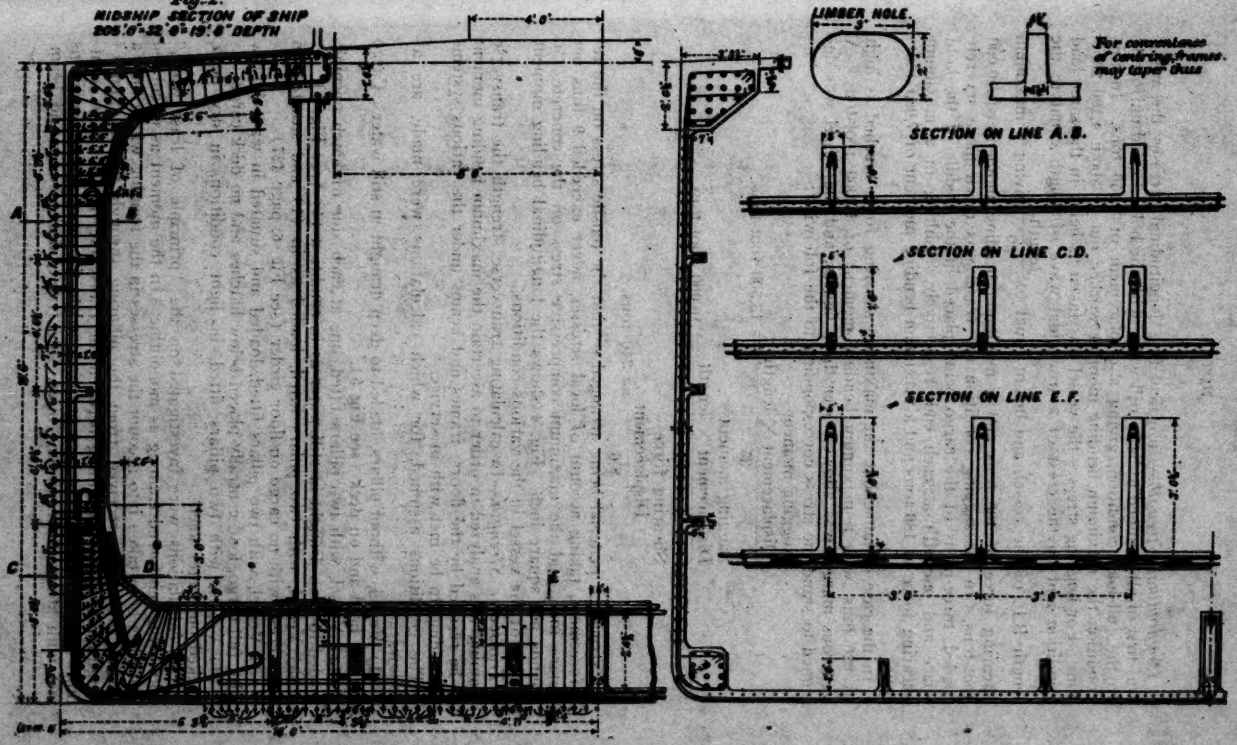


Fig.2.

MIDSHIP SECTION OF SHIP

Fig. 2.
MIDSHIP SECTION OF SHIP
305'-0"-32'-0"-19'-6" DEPTH



Longitudinal Strength.—In calculating longitudinal stresses the vessel was first of all assumed to be in still water with the holds loaded uniformly to the full-load condition, giving a displacement of 2,350 tons. In this condition the maximum bending moment worked out at about 4,100 foot-tons. For hogging stresses the vessel was taken as being in the same load condition and on the crest of a trochoidal wave of length equal to the length B.P. of the vessel, and height equal to $\frac{\text{length}}{20}$ giving a maximum shearing force of 180 tons and a maximum bending moment of 10,000 foot-tons. In the sagging condition the vessel was much more severely loaded, three-fifths of the cargo being placed in the middle of the hold space and one-fifth at each end, respectively, resulting in a maximum shearing force of 180 tons and a maximum bending moment of 7,000 foot-tons.

In the case of launching, the maximum shearing force reached as high as 220 tons, with a maximum bending moment of 10,000 foot-tons. A generous margin was allowed over these figures, and the vessel was designed to stand the stress corresponding to the following:

Hogging moment—

$$\frac{\text{Displacement} \times \text{length}}{35} = 13{,}500 \text{ foot-tons.}$$

Sagging moment

$$\frac{\text{Displacement} \times \text{length}}{48} = 10{,}000 \text{ foot-tons.}$$

Shearing force—

$$\frac{\text{Displacement}}{8{\cdot}6} = 240 \text{ tons.}$$

the structural material being arranged so that the tensile stress on the reinforcement, taking account of local stresses, never exceeded 9 tons per square inch, and the maximum compressive stress on the concrete 750 pounds per square inch. Fig. 4 shows the longitudinal bending moment curves of the vessel in the various conditions.

Transverse Strength.—In calculating transverse strength the transverse framing was analyzed in order to ascertain the maximum bending moment to be resisted by the floors, frames and beams under the various systems of loading to be met with in service.

The conditions assumed, for which calculations were made, are as follows:

(1) Vessel, without pillars, loaded to deep draught in still water. Cargo load in hold and on deck (see Fig 5).

(1*a*) Vessel, with two pillars fitted, one at each side of hatch opening, loaded to deep draught in still water as in condition (1) (see Fig. 5).

(2) Vessel, with two pillars fitted, loaded as in (1) and situated on wave crest, with no cargo on floor girder (see Fig. 6, page 427).

(3) Vessel, with two pillars fitted, loaded and situated in wave hollow with full cargo, load centrally placed below hatches and no deck cargo.

(4) Vessel, with two pillars fitted, in light condition, in dry dock, docked on center keel.

The calculations were investigated on the "principle of least work," taking the reinforced framing as monolithic. In the moment as calculated there has been taken into account the stresses in the framing where only a partial support is received from the pillaring through the medium of longitudinal girders spanning between the transverse frames. The scantlings of the framing have been proportioned to the more severe conditions met with at the floor girders, bilge, side frame, deck corner and beam.

CONCRETE CARGO VESSEL OF 1,150 TONS DEADWEIGHT
LINES PLAN
Fig. 3.

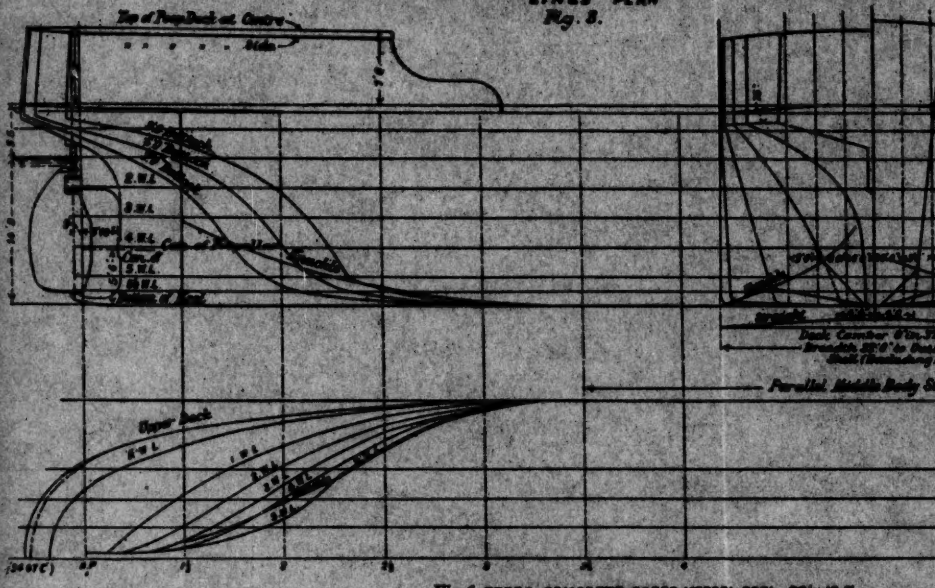
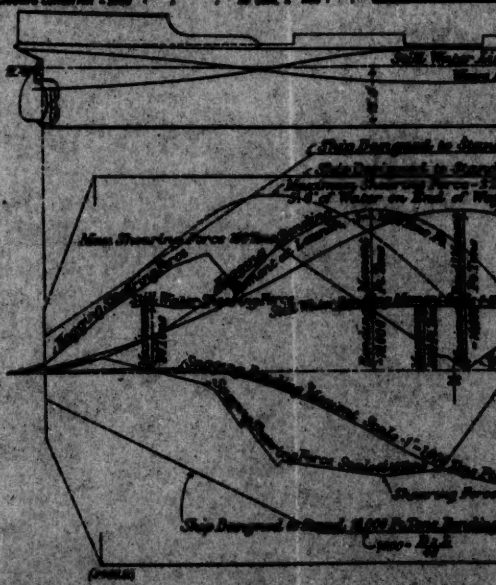
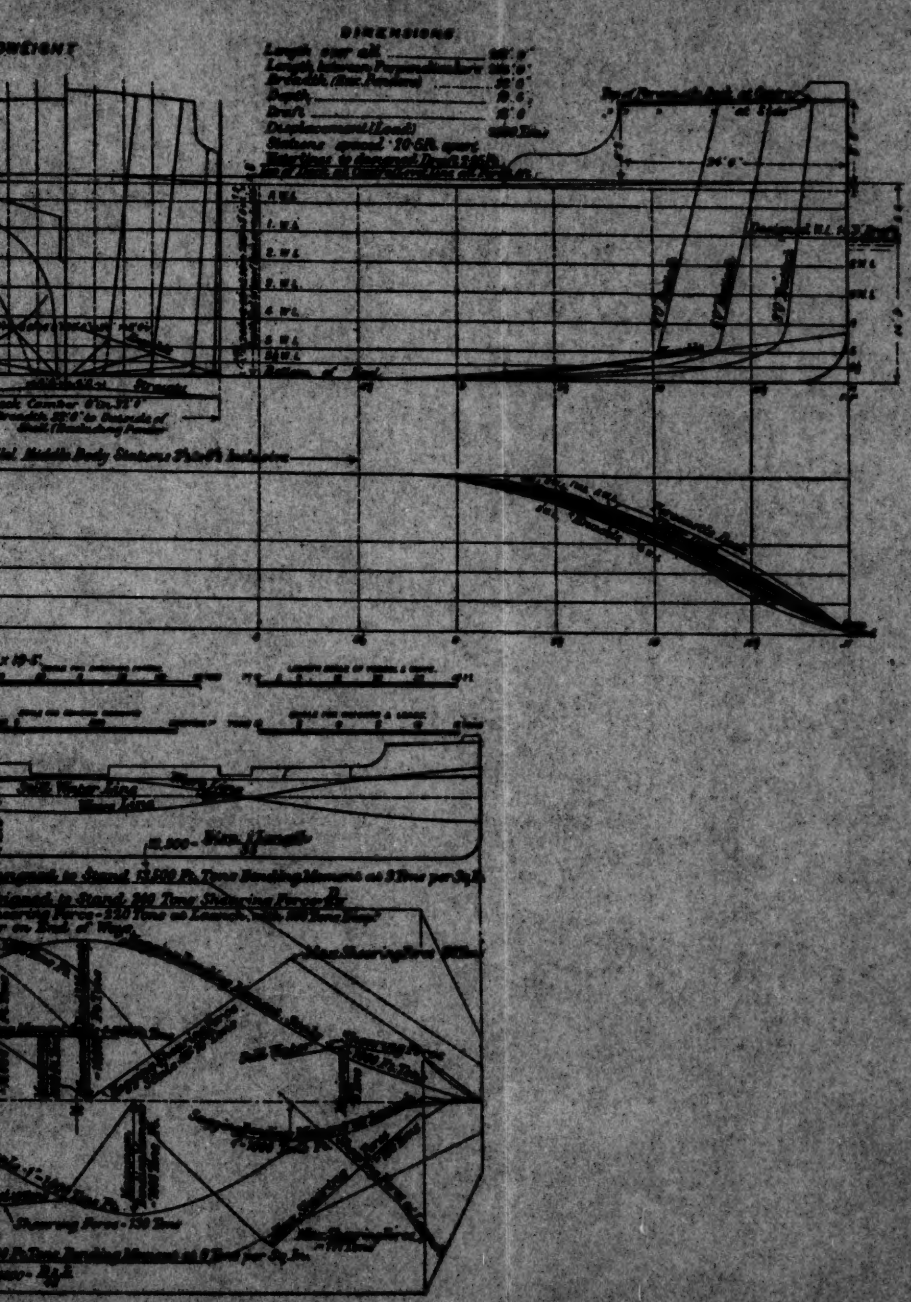
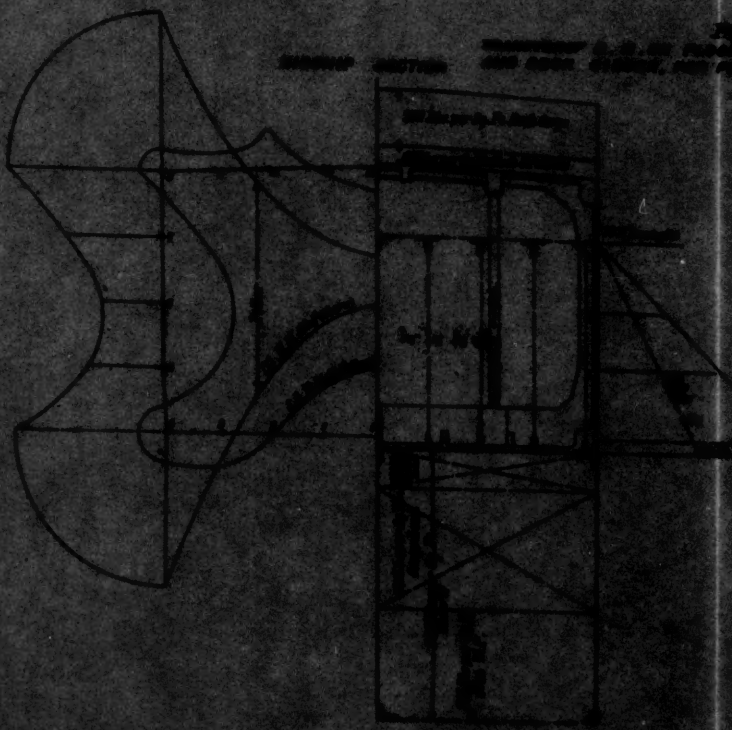
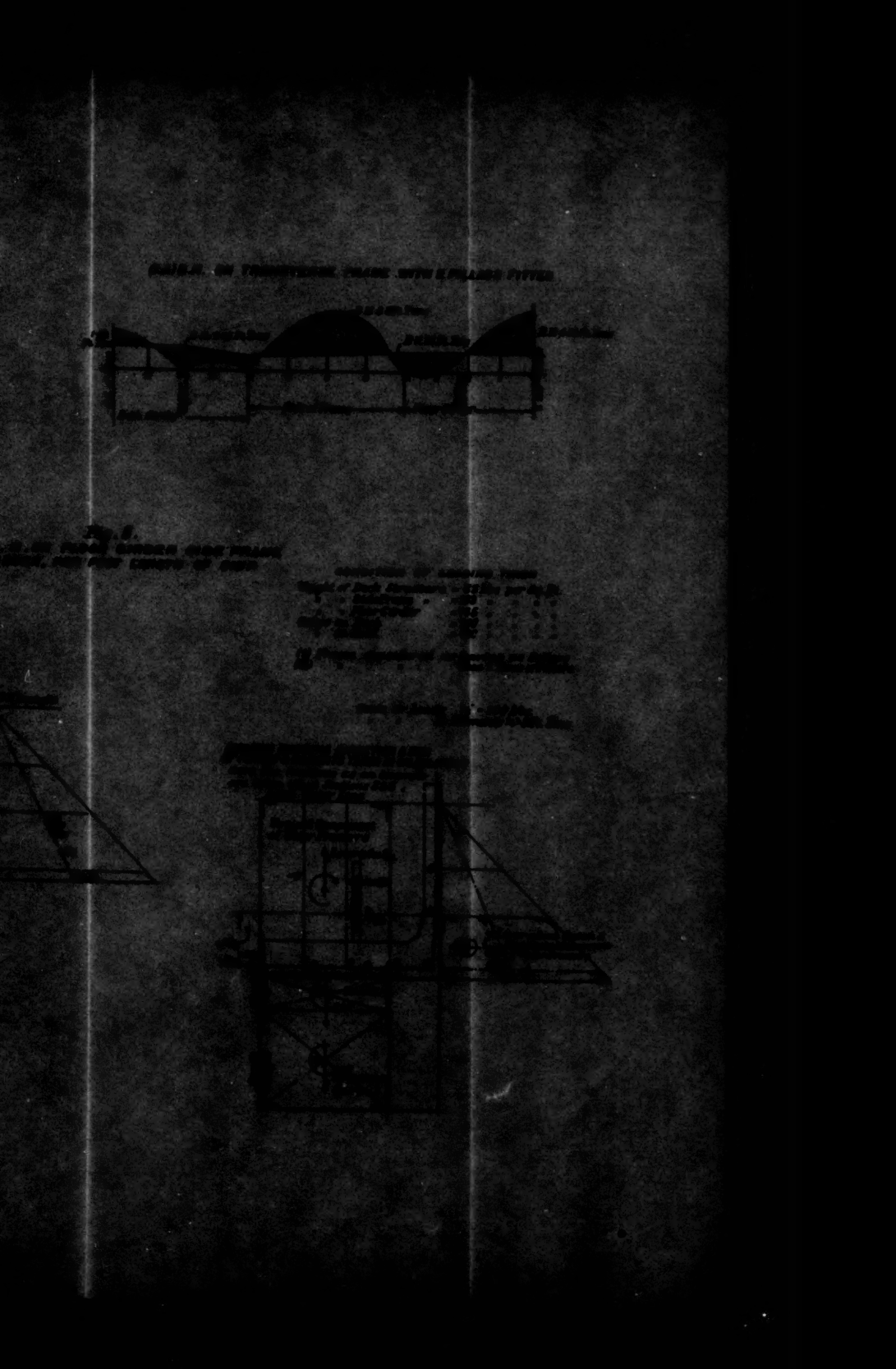


Fig 4. FERRO CONCRETE CARGO VESSEL 250' x 32' x 19-6"
CURVES OF BENDING & SHEARING STRESSES
HEIGHT OF WAVE = 1/20 LENGTH.




DIMENSIONS
Length over all
Length between Perpendiculars
Beam, Moulded
Depth
Draft
Displacement (Loads)
Stations spaced 10 Ft. apart
Water Lines
Bottom of Keel
Middle Body Stations Shaft Brackets
Steering Force - 150 Tons





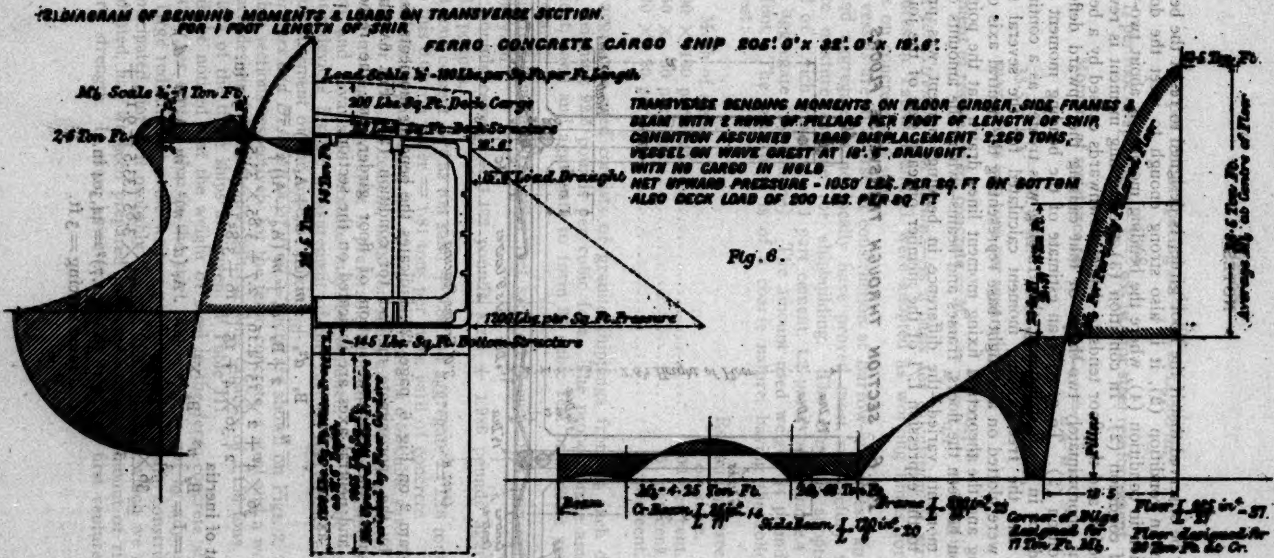

(2) DIAGRAM OF BENDING MOMENTS & LOADS ON TRANSVERSE SECTION
FOR 1 FOOT LENGTH OF SHIP
FERRO CONCRETE CARGO SHIP 205'.0" x 32'.0" x 19'.6".
Load Scale ⅛" = 100 Lbs. per Sq. Ft. per Ft. Length
200 Lbs. Sq. Ft. Deck Cargo
Mb Scale ⅛" = 1 Ton Ft.
2·6 Ton Ft.
W. L.
1200 Lbs. per Sq. Ft. Pressure
145 Lbs. Sq. Ft. Bottom Structure
TRANSVERSE BENDING MOMENTS ON FLOOR GIRDER, SIDE FRAMES & BEAM WITH 2 ROWS OF PILLARS PER FOOT OF LENGTH OF SHIP
CONDITION ASSUMED LOAD DISPLACEMENT 2,250 TONS,
VESSEL ON WAVE CREST AT 16'.6" DRAUGHT.
WITH NO CARGO IN HOLD
NET UPWARD PRESSURE - 1050 LBS. PER SQ. FT ON BOTTOM
ALSO DECK LOAD OF 200 LBS. PER SQ. FT
Average Mb 6 Ton Ft. on Centre of Floor
Pillar
18·5
Beam
Mb 4·25 Ton Ft.
Frame
Side Beam
Corner of Bilge designed for 17 Ton Ft. Mb.
Floor designed for 20 Ton Ft. do Cr.

Fig. 6.

It may be observed that if the floor girder is designed to resist the bending moment in condition (2), it is also strong enough to meet the docking condition in condition (4), where the bending moment is about two-thirds that of condition (2). In condition (3) the bending moment is reversed in direction and the floor tends to deflect downwards caused by a bending moment approximately two-thirds of that causing the upward deflection in condition (2). To provide an estimate of the bending moment to be allowed for in the first instance, the framing was treated as a continuous girder, and the free bending moment calculated for the several spans. These were plotted on a straight base representing the neutral axis of the framing and the theoretical fixing moment lines drawn; at the points of junction between the floors, frames and beams—where the amounts of the fixing moments varied—the difference in bending moment was proportioned to the expression L/I of the member on each side of the joint.

Fig. 6A SECTION THROUGH TRANSVERSE FLOORS

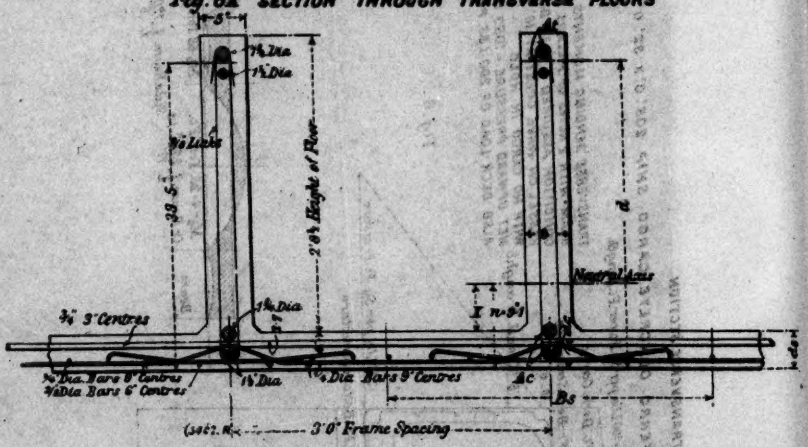


Diagram 2 on Fig. 6, page 427, indicates the bending moments as estimated by the approximate method for condition (2). Fig. 6A on this page shows the structural section of floor girder, and the respective moduli and neutral axes are indicated on the section.

Neutral axis above base

$$= n = \frac{B_s \cdot d^2_s + 2m\,(A_c \cdot d_c + A_t \cdot d)}{2\,\{B_s \cdot d_s + m\,(A_c + A_t)\}} =$$

$$\frac{36 \times 4^2 + 2 \times 15\,(4.16 \times 2.7 + 3.85 \times 33.5)}{2\,\{36 \times 4 + 15\,(4.16 + 3.85)\}} = 9.1 \text{ in.}$$

Moment of inertia

$$= I = \frac{B_s \cdot n^3}{3} - \frac{B_s \cdot X^3}{3} + m \cdot A_t\,(d-n)^2 + m \cdot A_c\,(n-d_c)^2$$

$$= \frac{36 \times 9.1^3}{3} - \frac{36 + 5.1^3}{3} + 15 \times 3.85\,(33.5 - 9.1)^2 +$$

$$15 \times 4.16\,(9.1 - 2.7)^2 = 44{,}304 \text{ in.}^4$$

Frame spacing = 3 ft.

$$\text{I per foot of spacing} = \frac{44,304}{3} = 14,768 \text{ in.}^4$$

$$\text{I}_s \text{ per foot of spacing} = \frac{14,768}{15} 985 \text{ in.}^4 \text{ for steel reinforcement.}$$

$$\text{Moment of resistance for steel} = M_{s.s.} = \frac{I_s}{d - n} =$$

$$\frac{985}{33.5 - 9.1} = 40.37 \text{ in.}^3 \quad \text{With B.M. of 30 ton-ft.,}$$

max. stress = 9 tons per square inch.

$$\text{Moment of resistance for concrete} = M_{c.c.} = \frac{I}{n} = \frac{14,768}{9.1} = 1,623 \text{ in.}^3$$

Basis for Strength Calculation.—As a basis for calculation of strength the figures in Tables I and II were adopted as working limits for safe stresses on concrete and steel.

The vessels are in course of construction at Barrow-in-Furness, and the sand and aggregate found locally have proved most satisfactory for a concrete mixture suitable for shipbuilding. The concrete adopted is formed of a rich mixture of 1 part cement, 1.2 parts sand, and 2.4 parts granite aggregate (by volume). The amount used weighs 84 pounds per cubic foot. The sand used is of a coarse nature, having the following sieve rest:

Mesh of sieve.	Remaining percentage on sieve.
100 × 100 per inch	100 per cent.
50 × 50 per inch	61 per cent.
20 × 20 per inch	14 per cent.

The aggregate used consists of granite chippings from ⅝ inch downwards, weighing 92.5 pounds per cubic foot. The proportionate amounts of cement, sand and aggregate to form 27 cubic feet or 1 cubic yard of concrete are:

9 cubic feet cement, 10.74 cubic feet sand, 21.47 cubic feet aggregate = 1 cubic yard.

= 757 pounds cement + 1.192 pounds sand + 1.986 pounds aggregate = 3.935 pounds.

= 0.34 tons cement + 0.53 ton sand + 0.89 ton aggregate = 1.76 tons.

Add to this 0.05 ton water = 1.81 tons per cubic yard of concrete.

For estimating purposes 1 ton of concrete contains 0.57 cubic yard = 15.4 cubic feet; or 0.192 ton cement, 0.300 ton sand, and 0.508 ton granite.

Machine mixing of concrete is preferable to hand mixing inasmuch as a standard and invariable mixture may be obtained, due to the use of automatic devices for the measurement of the various ingredients as well as of measurement of the time of mixing.

Particulars and Description of Hull.—There are shown on Figs. 2, 7 and 8, midship sections respectively for the ferro-concrete, a steel and a wooden ship, each designed for a deadweight carrying capacity of 1,150 tons. The particulars of the three ships are given in Table III.

It will be noticed that the weight of steel in the ferro-concrete ship is about 42½ per cent of that in the steel ship. This is not by any means such a large reduction as some writers on ferro-concrete ship construction have anticipated, nor is it probably the minimum quantity which we could have adopted, had we been prepared to incur a certain amount of risk in structural strength. In view, however, of this being our first venture in a

TABLE I.—BASIS FOR CALCULATION OF STRENGTH FOR CONCRETE.

Item.	Working stress.	Ultimate stress.
	lb. per sq. in.	lb. per sq. in.
Compression in beams	750	4,000
Compression direct	700	4,000
Shear	70	280
Adhesion	60	240
	100	500

TABLE II.—BASIS FOR CALCULATION OF STRENGTH FOR STEEL.

Item.	Working stress.	Elastic limit.	Ultimate stress.
	tons per sq. in.	tons per sq. in.	tons.
Tension, spiral bars	9	23½	30 to 36
Tension, plain bars	7½	15	27 to 30
Compression	7½	15	27 to 30
Shear	5½	...	24

TABLE III.—DIMENSIONS OF REINFORCED CONCRETE, STEEL AND WOOD SHIP OF 1,150 TONS DEADWEIGHT CARRYING CAPACITY.

...	Reinforced concrete.	Steel.	Wood.
	ft. in.	ft. in.	ft. in.
Length	205 0	188 0	205 0
Breadth	32 0	30 3	36 0
Depth	19 6	17 3	18 9
Draught	15 6	14 6	16 6*
	tons.	tons.	tons.
Deadweight	1,150	1,150	1,150
Displacement (load)	2,350	1,800	2,400
Indicated horse-power (about)	400	400	400

*Includes 12-inch wood seal.

...	Weights.		
	Reinforced concrete.	Steel.	Wood.
	tons.	tons.	tons.
Steel in hull†	190	445	140
Remainder of hull	860	80	960
Machinery and boilers	80	80	80
Outfit‡	70	45	70
Deadweight	1,150	1,150	1,150
Load displacement	2,350	1,800	2,400

†Includes hull castings and forgings.
‡Includes anchors, cables, boats, auxiliary machinery, etc.

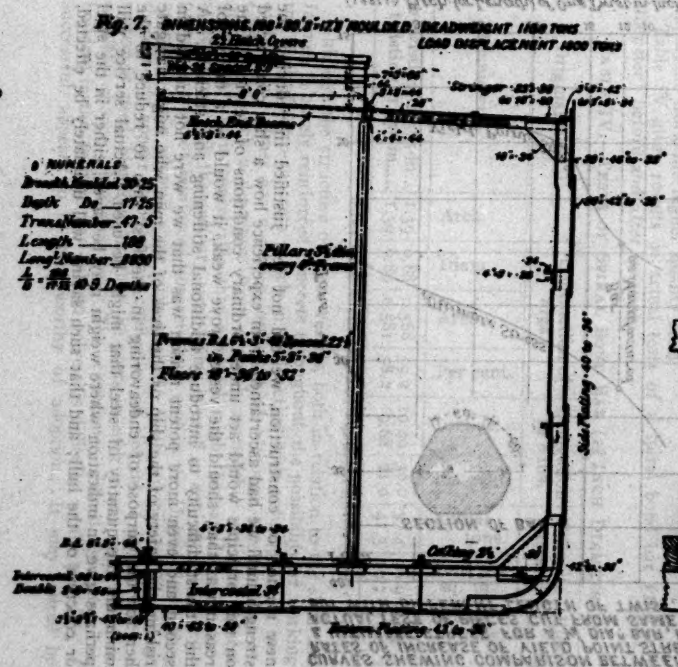
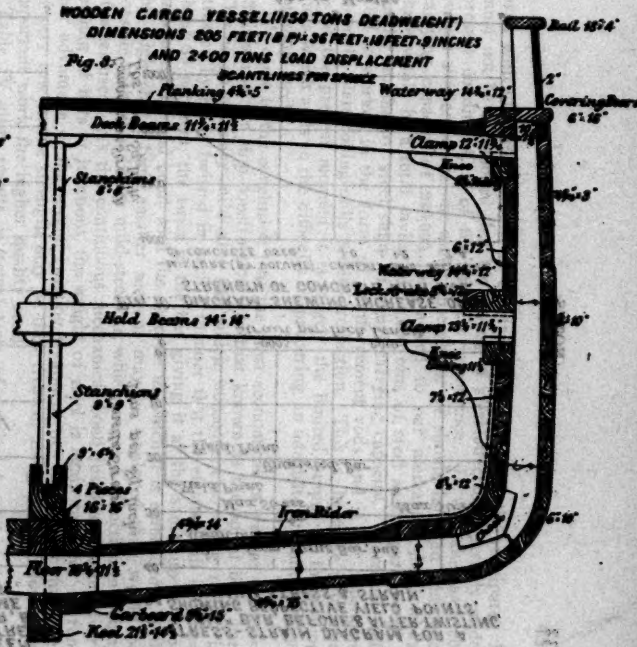
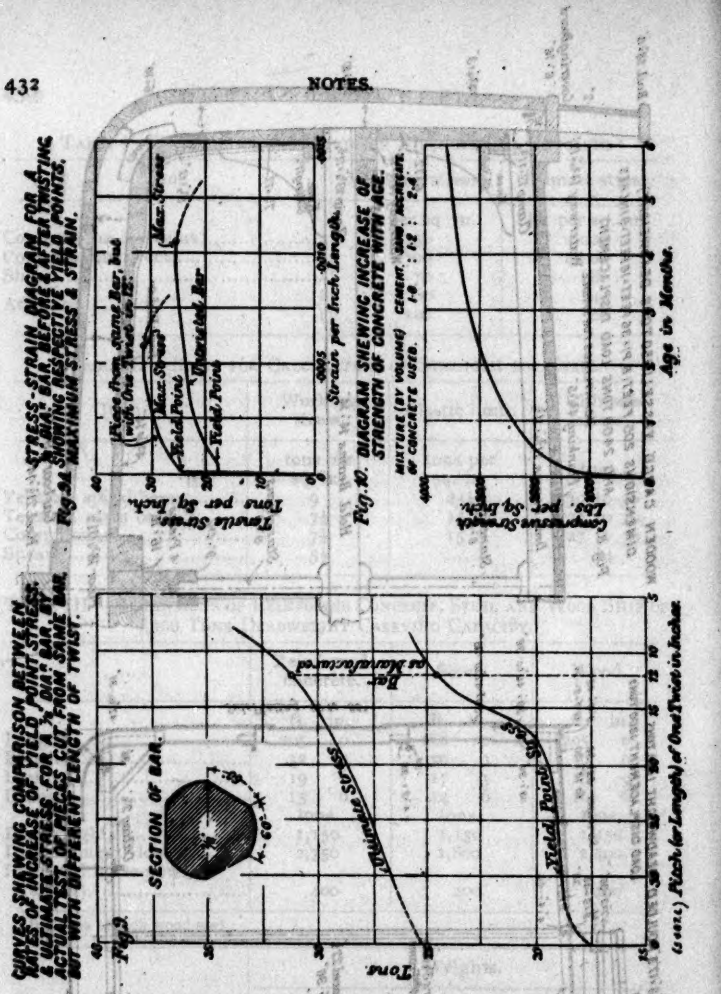


Fig. 7.
DIMENSIONS 100 x 25'3" x 12'3" MOULDED. DEADWEIGHT 1150 TONS
LOAD DISPLACEMENT 1600 TONS
Pillars 5" dia. every 8' centres
Floors 18'-36" to 52"
Intercostal 28"
Bottom Plating 42" to 52"
Side Plating 40 to 36"
Ceiling 5"


WOODEN CARGO VESSEL (1150 TONS DEADWEIGHT)
DIMENSIONS 205 FEET (B P) x 36 FEET x 18 FEET 9 INCHES
AND 2400 TONS LOAD DISPLACEMENT
Fig. 8.
SCANTLINGS FOR FRAMES
Planking 4½"x5"
Waterway 14½"x12"
Deck Beams 11½"x11½"
Clamp 12"x11½"
Stanchions 8"x8"
Hold Beams 14"x14"
Clamp 13"x11½"
Stanchions 8"x8"
4 Pillars 16"x16"
Iron Ridder
Floor 10½"x11½"
Garboard 8½"x15"
Keel 21½"x14½"


STRESS-STRAIN DIAGRAM FOR A ½" DIA. BAR, BEFORE & AFTER TWISTING. SHOWING RESPECTIVE YIELD POINTS, MAXIMUM STRESS & STRAIN.
Tensile Stress Tons per Sq. Inch.
Strain per Inch Length.
Max. Stress
Yield Point
Thoroughly Bar
Fig. 10. DIAGRAM SHEWING INCREASE OF STRENGTH OF CONCRETE WITH AGE.
MIXTURE (BY VOLUME) OF CONCRETE USED. CEMENT: SAND: AGGREGATE. 1:2:4
Compressive Strength Lbs. per Sq. Inch.
Age in Months.
CURVES SHEWING COMPARISON BETWEEN RATES OF INCREASE OF YIELD POINT STRESS & ULTIMATE STRESS, FOR A ½" DIA. BAR, BY ACTUAL TEST OF PIECES CUT FROM SAME BAR, BUT WITH DIFFERENT LENGTH OF TWIST.
Fig. 9.
SECTION OF BAR.
Bar as Manufactured
Ultimate Stress
Yield Point
Tons.
(Length of Pitch for Length) of One Twist in Inches.

new method of construction, we did not feel justified in reducing the strength until we had ascertained from experience how a ship constructed on this principle would act under ordinary conditions of service. One reason was that, should the vessel prove weak, it would be a matter of the utmost difficulty to introduce additional stiffening and strength. A second and even more potent reason was that we were not justified in risking the safety of the ship or the lives of the men who might navigate her for the purpose of endeavoring in the first ship to reduce to the minimum the quantity of steel that might be used. Actual service will perhaps give an indication where weight may be saved, either in the steel or concrete of the hull; and that such saving will ultimately be effected,

even with identical methods of construction, we confidently anticipate. It must not be overlooked that the saving of steel effected, especially under present conditions, is of the greatest importance. If we take, say, 800 vessels similar to the one under discussion, we should have an additional carrying capacity of 230,000 tons added to our mercantile fleet, whilst there would be a total saving of 51,000 tons of steel as compared with a similar number of steel ships of equal carrying capacity. Moreover, this steel is worked in the form of reinforcement rods and is not dependent upon the plate-rolling mills for its production. This is the more important as at the present moment it is largely the limited output of the rolling mills which adds to the difficulty of procuring an adequate amount of ship-building steel.

We came to the conclusion that the most economical as well as the most efficient rod for ship construction was that known as the spiral bond bar. This is an ordinary mild-steel bar which, subsequent to being rolled, is stressed by being twisted in such a manner that the elastic limit is raised about 35 per cent, the twist in the bar giving it at the same time a continuous mechanical bond with the concrete in which it is embedded. With this high elastic limit smaller sections may be employed than would be possible with an ordinary mild-steel bar, which means a reduction in weight of steel used. As the continuous mechanical bond justified shorter overlapping of bars at junctions, the weight of steel is again reduced. A further advantage of the special twisting treatment lies in the fact that steel of very low tensile strength may be thus treated, and so release for other important work the higher quality steel.

Table IV gives results of actual tests of a spiral bond bar (1⅜ inch diameter) before and after twisting.

TABLE IV.—TESTS OF SPIRAL BOND BAR (1⅜ INCH DIAMETER.)

Original size.		Contraction of area.			Elongation in 8 ins.		Ultimate stress.		Elastic limit.	
Diameter.	Area.	Diam.	Area.	Per cent.	Inches.	Per cent.	Actual tons.	Tons per sq. in.	Actual tons.	Tons per sq. in.
Before twisting, 1.375 in...	1.39	0.88	0.568	59.2	10.40	30.0	37.4	26.9	24.5	17.62
After twisting, 1.375 in.....	1.39	0.89	0.581	58.2	9.68	21.0	41.3	29.7	33.2	23.86

See stress strain diagram, Fig. 9.

Although the structure of these vessels, both in relation to the shell and frames, is purely reinforced concrete throughout, the possibility of working the shell of ferro-concrete vessels in conjunction with ordinary frame bars or built-up frames has not been lost sight of, although the problems attendant on such a departure from a purely monolithic structure present difficulties. It is conceivable that a vessel might be satisfactorily constructed in which the ordinary steel frame is used in conjunction with the present system of reinforced concrete hull, and in the United States they appear to have adopted some such method. With our present knowledge there would appear to be in this connection an element of danger unless a system is used, blending together the parts in such a way as to leave no possible chance of disruption. The experience gained by American builders in this respect may help to solve this difficulty, as the principle, if successful, appears to be well worthy of adoption. It may indeed be

essential in ferro-concrete ships of large tonnage. Such a system, too, would overcome some of the difficulties in connection with the fitting of various details, which, at present, it is necessary to arrange for before the concrete portion is commenced. A further advantage, and probably the principal one, is that without having to introduce a much greater proportion of skilled labor, the system would result in a decreased weight of hull.

Hull Fittings and Their Connection.—Many problems have arisen regarding fittings and their connection and relation to the ship, which in ordinary steel construction are simple, but which, in the case of ferro-concrete construction, offer difficulties which have had to be carefully examined and overcome. Amongst these are the openings in the ship's bottom, stuffing-boxes, the passage of pipes through bulkheads, the connection of fittings to the bulkheads, and the connection of stanchions, fairleads, bollards, etc., to the deck. Added to these are important problems, such as the connection to the hull of the rudder post, stern post, and the stern tube. In order to illustrate some of the difficulties experienced and some of the means adopted to overcome these difficulties, I have appended to this paper some sketches showing how certain of these questions have been dealt with, such as, for instance:

Fig. 11—Arrangement of pillaring.

Fig. 12, on folder with Fig. 15.—Typical spigot for sea connections.

Fig. 13—Ceiling plan.

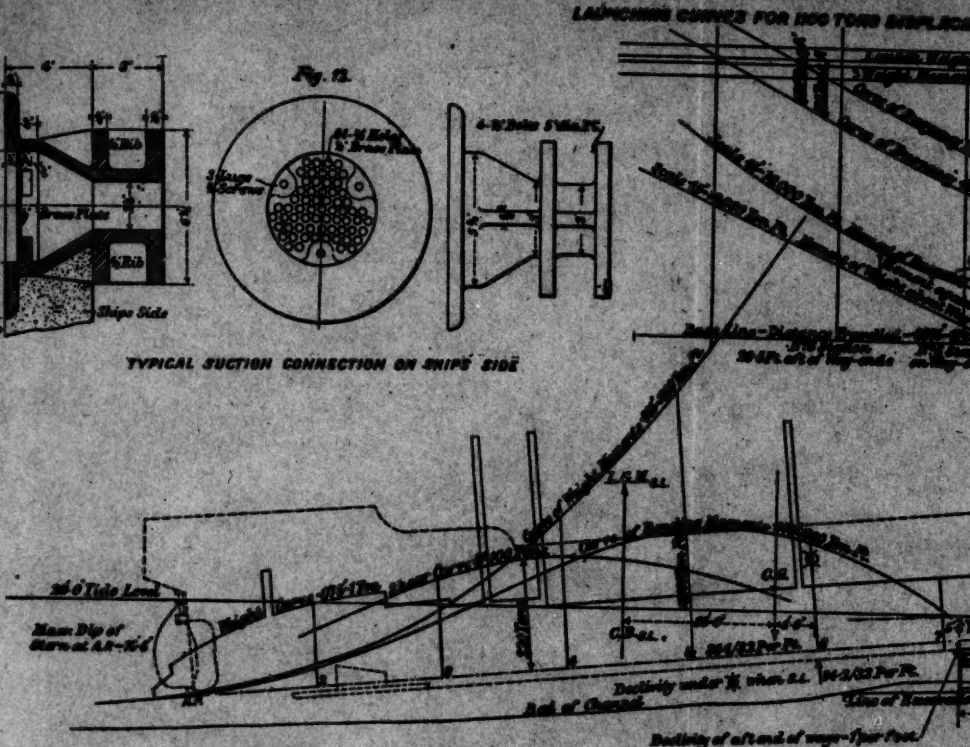
Fig. 16—Stern frame.

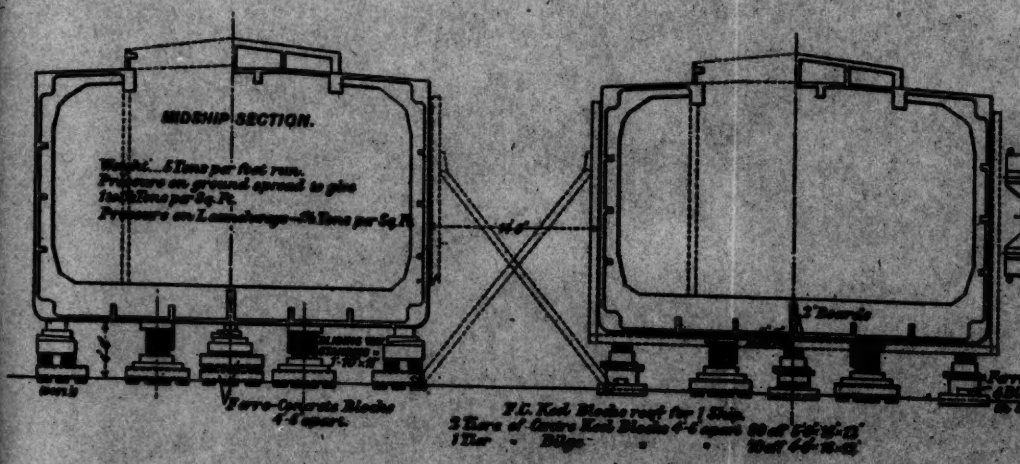
There is no doubt that, with experience, improved methods of adapting fittings and parts of the structure to their proper functions in economical and efficient manner will be evolved, but for the moment we think that the methods adopted, as illusrated in the drawings reproduced, are such as to give reasonable justification for anticipating efficient performance. In the case of a steel ship, if before the erection of the bulkhead provision has not been made and holes punched to take any particular fitting, as, for instance, a bracket, it is an easy matter to drill such holes afterwards to accommodate it. For a concrete vessel, however, the cutting of holes after the concrete has set is not easy and is undesirable. It is, therefore, highly important that all the details of fittings required to be connected to the hull, or passing through a bulkhead, for instance, should be considered before the erection and casting of that bulkhead, and the necessary information must be ready at an early stage of the construction in order that the builders may make suitable provision for the same when erecting the shuttering. Where it is necessary to leave a hole in the concrete, a wooden peg may be fastened to the shuttering in the required position; but in the case of pipes passing through a deck or a bulkhead, a better arrangement is to cast a metal spigot in the concrete with suitable flanges or screwed ends projecting on each side for connection to the pipes. Such fittings as mooring pipes or hawse pipes, as well as the majority of small topside fittings, have to be specially designed, with proper projections, for casting into the concrete, as otherwise holes must be prepared to take the holding-down bolts.

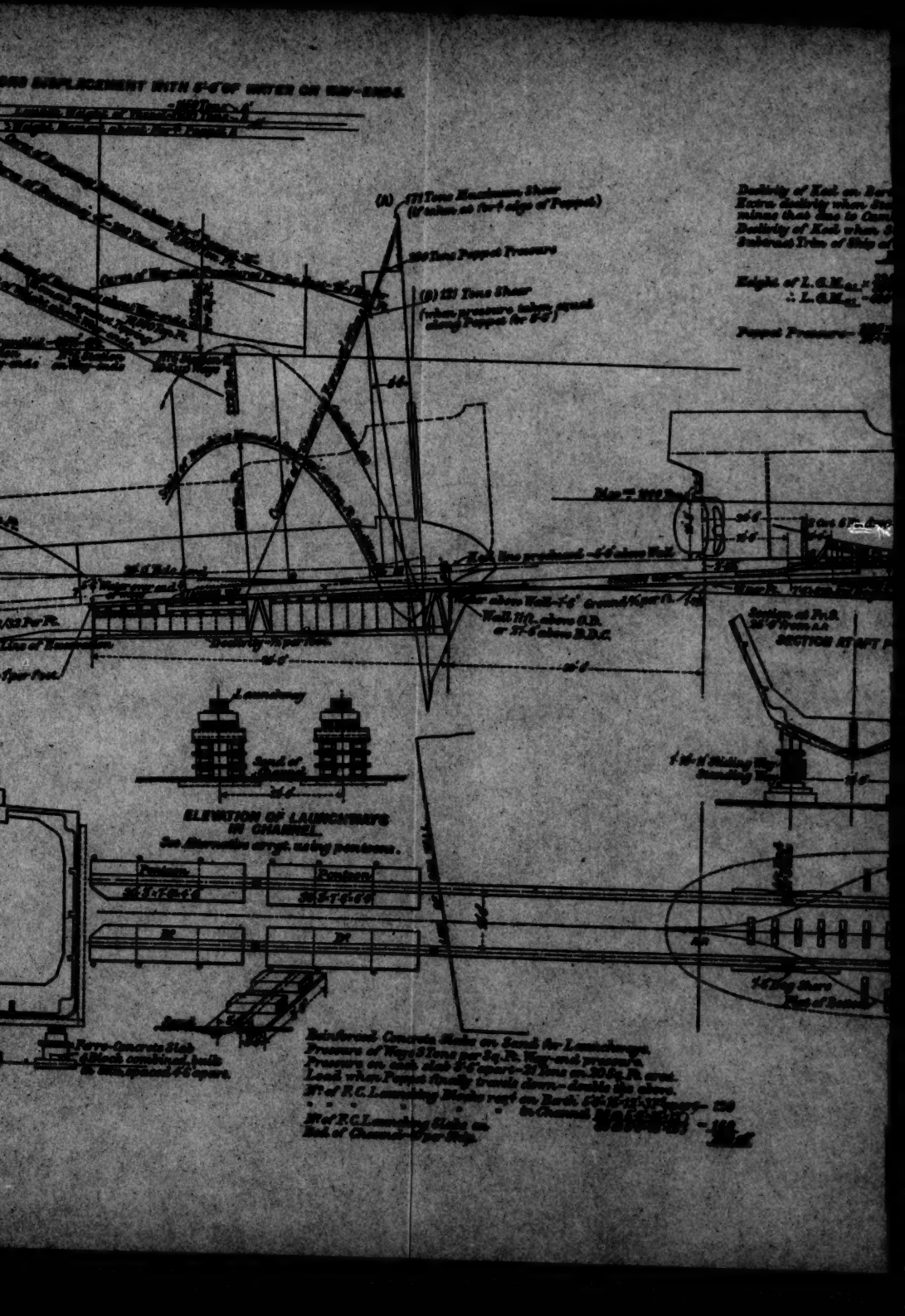
Lines and Tank Trials.—It is obvious, in view of the fact that the whole of the vessel's hull has to be cast in molds or shutters, that the simpler and straighter the lines of the vessel the more cheaply and easily the shuttering can be constructed, and the more rapidly the work can be proceeded with. For this reason, when designing the lines, a simple midship section was adopted having a perfectly straight side and bottom with only a very small curvature at the bilge. This section was retained in a parallel middle

XEROX

TYPICAL SUCTION CONNECTION ON SHIPS SIDE


LAUNCHING CURVES FOR 1000 TONS DISPLACE
Base Line—Diagonal
Depth of Water Required
Declivity of aft end of ways—1 per Foot.


MIDSHIP SECTION.
Weight—5 Tons per foot run.
Pressure on ground spread to give
1 to 1½ Tons per Sq. Ft.
Pressure on Launchways—6½ Tons per Sq. Ft.
Keel Blocks rest for 1 Ship.
Bilge
Ways


(A) 771 Tons Maximum Shear
(When on fore edge of Poppet)
350 Tons Poppet Pressure
(B) 121 Tons Shear
(when pressure taken equal along Poppet for G.C.)
Max. 1002 Tons
Wall Top raised -6'-0" above Wall
Wall Top above Wall -6'-0' Ground Level
or 27-6 above B.D.C.
Section at Fr. 0
26'-0" from A.P.
SECTION AT AFT
ELEVATION OF LAUNCHWAYS IN CHANNEL
See Alternative arrgt. using pontoons.
Pontoon
36 ST.T.C.C.
Pontoon
36 ST.T.C.C.
Ferro-Concrete Slab
4 Block combined both
to 30 ft. spaced 4'-0 apart.
Reinforced Concrete Blocks on Sand for Launchways
Pressure of Ways 3 Tons per Sq. Ft. Way and pressure
Pressure on each slab 3'-0 apart=27 Tons on 20 Sq. Ft. area
Load when Poppet finally travels down=double the above.
W. of R.C.I. Launching Blocks rest on Earth
W. of R.C.I. Launching Blocks on
Bed of Channel

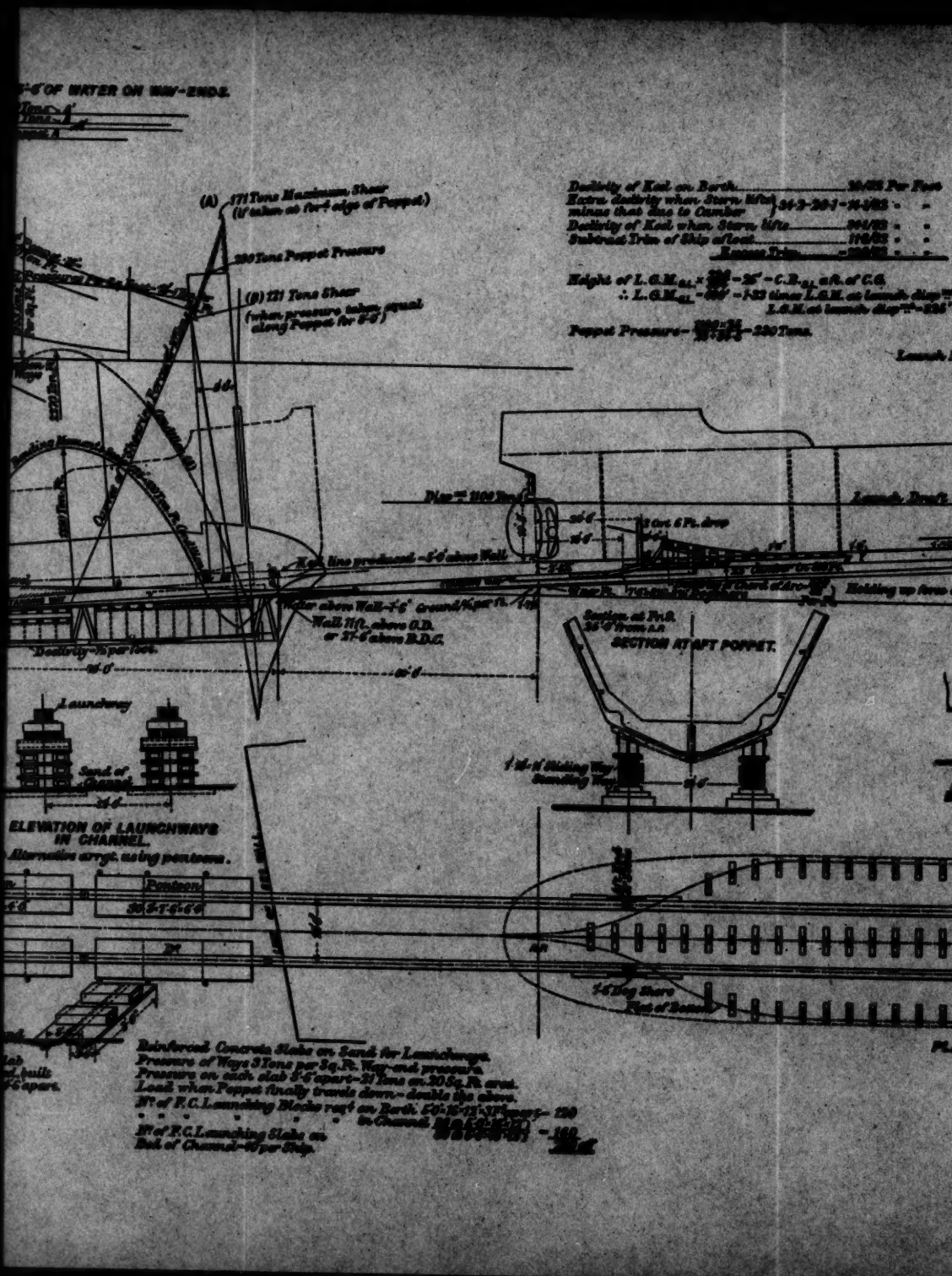
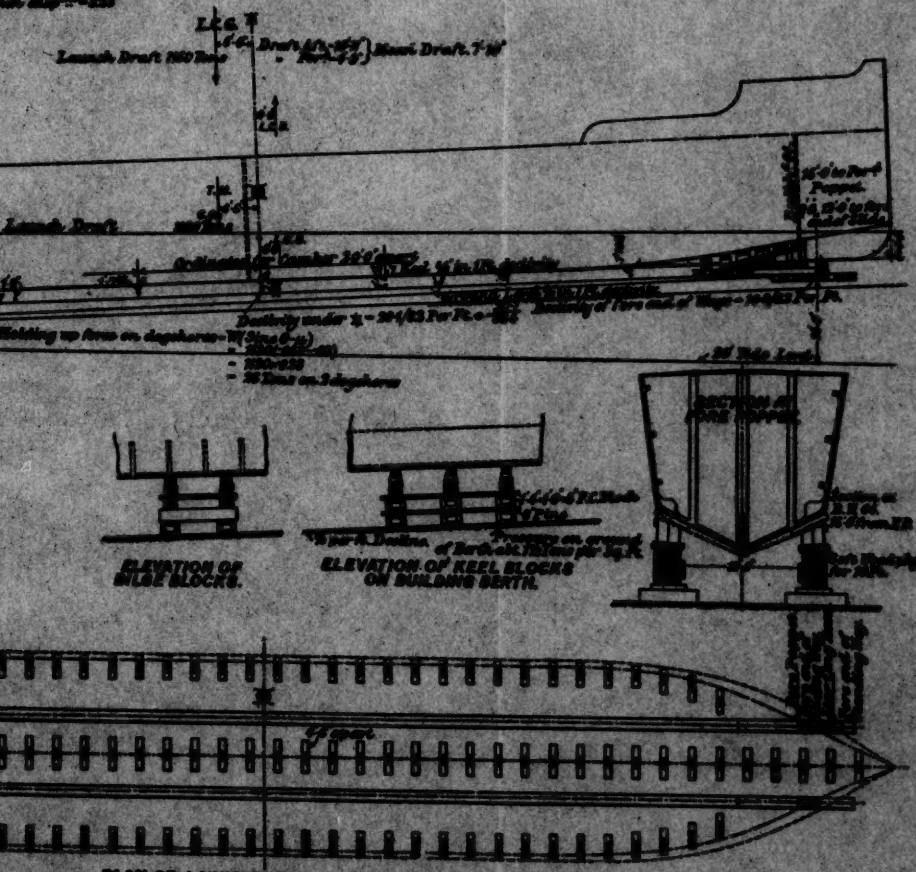

(A) 171 Tons Maximum Shear
(If taken as for 4 legs of Poppet)
230 Tons Poppet Pressure
(B) 121 Tons Shear
(when pressure taken equal along Poppet for 5'-0")
Water above Wall-4'-6" Ground Kept Fr.
Wall lift above O.D. or 27'-6" above B.D.C.
SECTION AT AFT POPPET.
Launchway
Sand of Channel
ELEVATION OF LAUNCHWAYS IN CHANNEL.
Alternative arrgt. using pontoons.
Pontoon
Reinforced Concrete Slabs on Sand for Launchways.
Pressure of Ways 3 Tons per Sq. Ft. Way-end pressure
Pressure on each slab 5'-0" apart-21 Tons on 20 Sq. Ft. area.
Load when Poppet finally travels down-double the above.
No of P.C.L. sanding Blocks rqd on Berth.
No of P.C.L. sanding Slabs on Bed of Channel-46 per Ship.

Fig. 15.

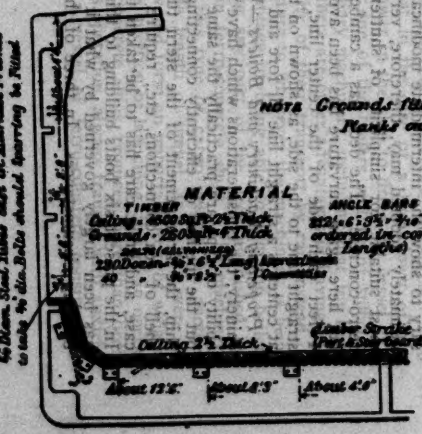


ELEVATION OF SIDE BLOCKS.

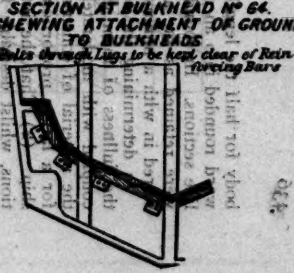
ELEVATION OF KEEL BLOCKS ON BUILDING BERTH.

PLAN OF LAUNCHWAYS AND KEEL BLOCKS.

Fig. 18.


MATERIAL
TIMBER
Ceiling - 4600 Sq Ft - 2" Thick.
Grounds - 250 Lin Ft - 1" Thick.
230 Doz - ½" x 6" Long
40 " ½" x 2½" "
ANGLE BARS
212 - 6" x 3½" x ⅜" to be ordered in convenient Length(s)
Ceiling 2" Thick.
Timber Stroke (Port & Stbd Coast)
About 12'6"
About 4'3"
About 4'0"

SECTION AT BULKHEADS Nos 20&24
SHEWING ATTACHMENT OF GROUNDS TO BULKHEADS
Bolts Through Lugs to Floors & Bulkheads to be kept clear of Reinforcing Bars & Stoke


SECTION AT BULKHEAD No 64.
SHEWING ATTACHMENT OF GROUNDS TO BULKHEADS
Bolts through Lugs to be kept clear of Reinforcing Bars

NOTE Grounds fitted on Alternate Floors throughout
Ranks only to be Bolted at ends

body for half the length of the vessel amidships, and the waterlines forward rounded into an easy entrance, still retaining, however, the straight-line sections. In rounding in the waterlines aft the straight-line sections were retained as far as possible and then run aft to the propeller and faired in with a minimum of curvature.

In determining the form two points had to be considered:—(1) That the fullness of form which is economical in ordinary cargo boats when coupled with moderately low speeds would certainly be desirable when the material of construction was such that the weight of hull accounted for a larger proportion of the displacement than in the case of a steel ship. (2) That the introduction of straight lines in the transverse sections, whilst desirable for constructional purposes, might lead to a large increase in resistance compared with an ordinary cargo-boat form, if there were any tendency for the stream lines to flow across the knuckle (*i. e.*, the intersection of the two surfaces forming the sides and bottom).

To obtain a reasonably good form under the above conditions, trials were carried out in the experimental tank as follows: The model was made fuller than was actually required and cast thick enough locally to allow cutting off at certain parts. After the curve of resistance had been obtained with the original model (C S), Fig. 14, the bottom part of the bow was fined (C S_2) with an improved result. Then the after body was similarly fined (C S_3) with further advantage. A further fining of the bow (C S_4) and then again of the stern (C S_5) still improved the form, which was now reasonably comparable with an ordinary cargo-boat form of the same dimensions and displacement. The total modifications made were comparatively slight, amounting to 23 tons, and reducing the block coefficient by .01.

The load waterline and curve of areas have generally been regarded as the determining feature of the resistance of an ordinary form, but we found that the influence of the knuckle was so marked that, with the waterline unaltered and practically the same curve of areas, a reduction was made in resistance of 40 per cent.

The lines plan, Fig. 3, folder shows in dotted lines the model as originally tried corresponding to the CS form on the tank results diagram. Fig. 14, and the full lines on the lines plan show the lines to which the ship is actually being built and correspond to the last trial in the tank and to the CS_5 form on the tank results diagram. It was not considered necessary to show the intermediate modifications on the lines plan. The lines ultimately adopted may, therefore, very reasonably be considered as the most suitable for simplicity of shuttering and for easy construction in ferro-concrete. The deck has a camber of 8 inches in the full breadth; but here again curvature has been avoided, the crown being a level flat 4 feet each side of the center line, the remaining deck sloping in a straight line to the side, as shown on the lines plan, Fig. 3, and the deck at center is a straight line all fore and aft.

Propelling Machinery and Boilers.—With respect to the propelling machinery, the considerations which have to be taken into account as to suitability of type are practically the same in ferro-concrete as in steel ships; but the method of efficiently connecting the machinery to the hull of the ship, the arrangement of the stern tubes, the special attachment to the shell of sea connections, etc., require more consideration in the former case, and great care has to be taken to ensure reliability in this respect. In the present six boats building to this design the adoption of machinery has been largely governed by what it has been possible to obtain under existing circumstances. In three of the vessels we are fitting compound

Fig. 14. TANK RESULTS
MODEL "C.S." (CONCRETE SHIP).
DIMENSIONS 282'0" × 35'0" × 16'0" MOULD.
ALL RUNS AT 2,560 TONS.
C.S.1 - Forefoot Fined.
C.S.2 - Fined Aft.
C.S.3 - Forefoot Further Fined.
C.S.4 - Aft End Further Fined.
C.S.1 & C.S.2 are for the Final Lines (See Body).
Effective Horse Power.
C.S.1
C.S.2
C.S.3
C.S.4
(Load. d.)
Knots

surface-condensing single-screw engines of approximately 350 H.P. to 400 H.P., with cylinders 17-inch and 34-inch diameter, 24-inch stroke working at about 100 r.p.m. The boiler installation consists of two cylindrical boilers 9 feet 6 inches in diameter by 9 feet long, working at a pressure of about 130 pounds. In the remaining three we intend fitting triple-expansion engines of about 500 I.H.P., which will somewhat reduce our deadweight capacity, but give an appreciable increase of speed. Two boilers were decided upon, so that in the case of a breakdown of one there would be sufficient boiler power in the ship to drive the vessel at a reasonable speed. When in port one of the boilers will be of sufficient power to work all necessary auxiliary machinery, including that for loading or discharging. Incidentally, there are other advantages in having two small boilers, such as the possibility of their transport by rail instead of by sea from the place of manufacture to wherever the machinery is being installed. There is, too, the advantage of being able to ship the boilers through the hatchways, as leaving any part of the deck unfinished for this purpose is very undesirable in ferro-concrete ships. The machinery is fitted aft, a usual position in this type of cargo vessel; but another reason for this was to reduce the length of tunnel, as if this were of ferro-concrete instead of steel it would form an important item of weight.

Launching.—Coming next to the question of launching, this is an operation which, in the case of ferro-concrete ships, requires more than ordinary consideration and care. This operation, always fraught with anxiety, is doubly so when dealing with a vessel of ferro-concrete construction.

Take, for instance, the launching arrangements for the vessel under discussion. A launching weight of 1,100 tons has to be dealt with as compared with about 550 tons in the case of a steel ship of corresponding size. To keep down the stresses, a moderate declivity of ways with ample depth of water on the way ends has been arranged; but, even so, quite a considerable hogging stress and excessive way-end pressures are met with just before the stern commences to lift. Some internal shoring has to be arranged at suitable parts of the vessel as additional precautions before launching. The shearing forces at the fore poppet are severe and must be met by an adequate strength of hull. The foundation supports, especially as the vessel is being built on new ground, have been carefully considered, as sinkage during any stage of the concreting would have serious results and must be guarded against. The precaution of allowing the concrete sufficient time to set properly before the vessel is launched is necessary in view of the fact that the compressive strength of the concrete continues to increase for some time after being worked, and it is necessary to utilize as much of this increased strength as possible to resist the launching stresses. Fig. 10, page 432, shows this increase of compressive strength on a base of months. The supporting blocks and holding-up arrangements (dog shores) also require greater strength than is necessary in the case of the lighter steel vessels. A sketch showing the arrangement made is given in Fig. 15, and shows very fully the practical arrangements as well as the curves of stresses under all conditions likely to be met with. Generally speaking, the severe launching conditions may be taken as an adequate test of the vessel's ability to withstand any stresses she may be called upon to meet under ordinary conditions of service.

LAUNCHING PARTICULARS.

Item.	
Length of standing ways	313 ft. + 6 ft.
Camber of 313 feet	22 ft.
Length and breadth of sliding ways	164 ft. × 23 in.
Area of sliding ways	600 sq. ft.
Launch displacement	1,100 tons.
Weight on standing ways	1,130 tons.
Pressure per square foot on ways	1.88 tons per sq. in.
C. G. aft of amidships	6.5 ft.
Poppet pressure	230 tons.
Moment against tipping	13,500 tons.
Bending moment on overhanging stern	10,000 ton ft.
Stress on concrete	350 lbs. per sq. in.
Tensile stress on reinforcement	5½ tons per sq. in.
Minimum water on way ends	5 ft. 6 in.

To obviate the undesirable necessity of pile-driving in the channel where the vessel is being built, the standing ways beyond the seawall are supported on specially designed pontoons resting on the bed of the channel. These pontoons are specially constructed of concrete and may be towed from one berth to another, and when in the necessary positions are sunk by being flooded through sluice valves arranged in the sides for that purpose (see Fig. 15).

Some statement should perhaps be made here as to the time for construction of vessels of this type as compared with steel ships, and our experience so far leads us to believe that in the case of the first vessel of any type, the time of construction approximates very closely to that of a

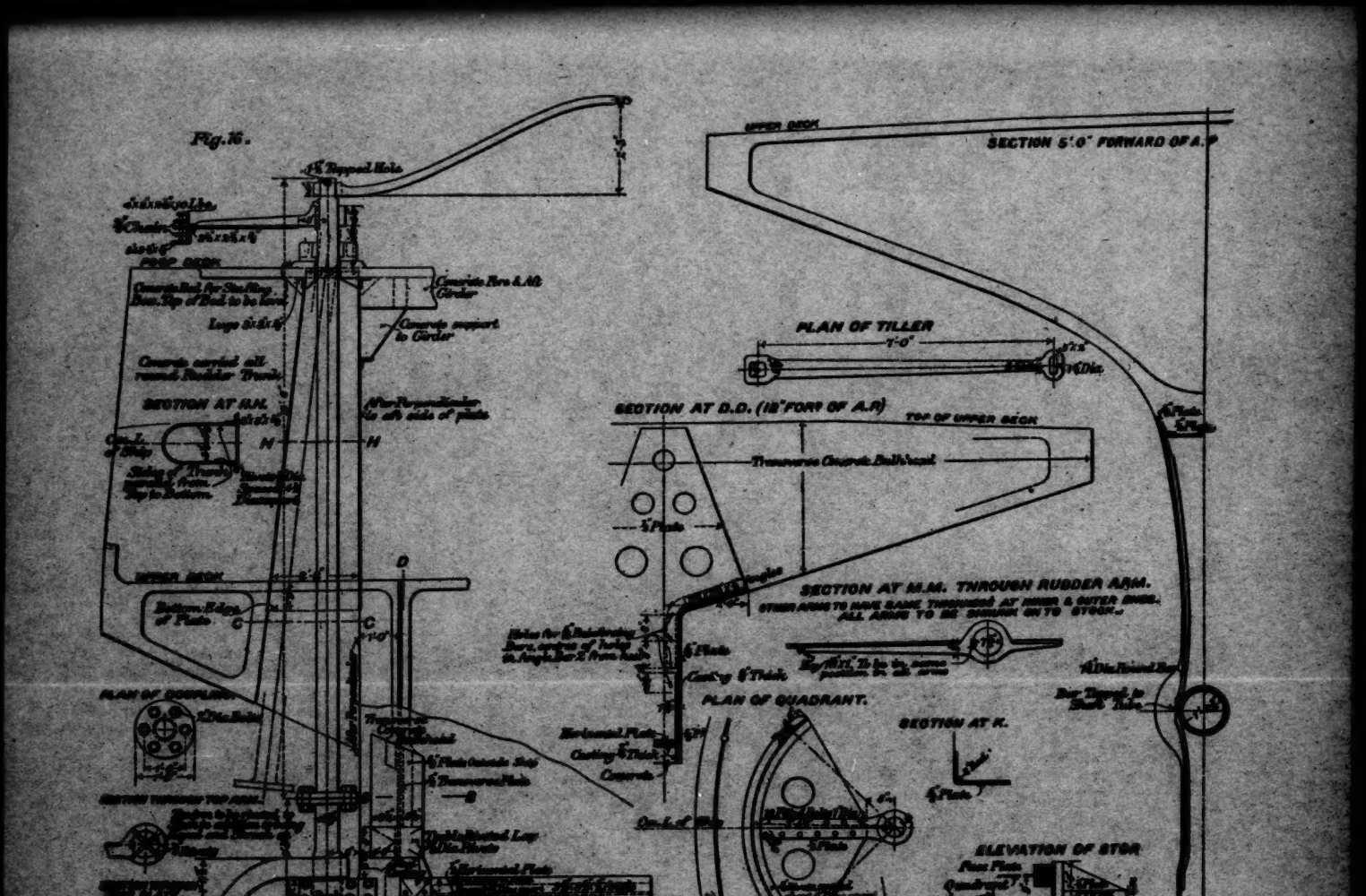
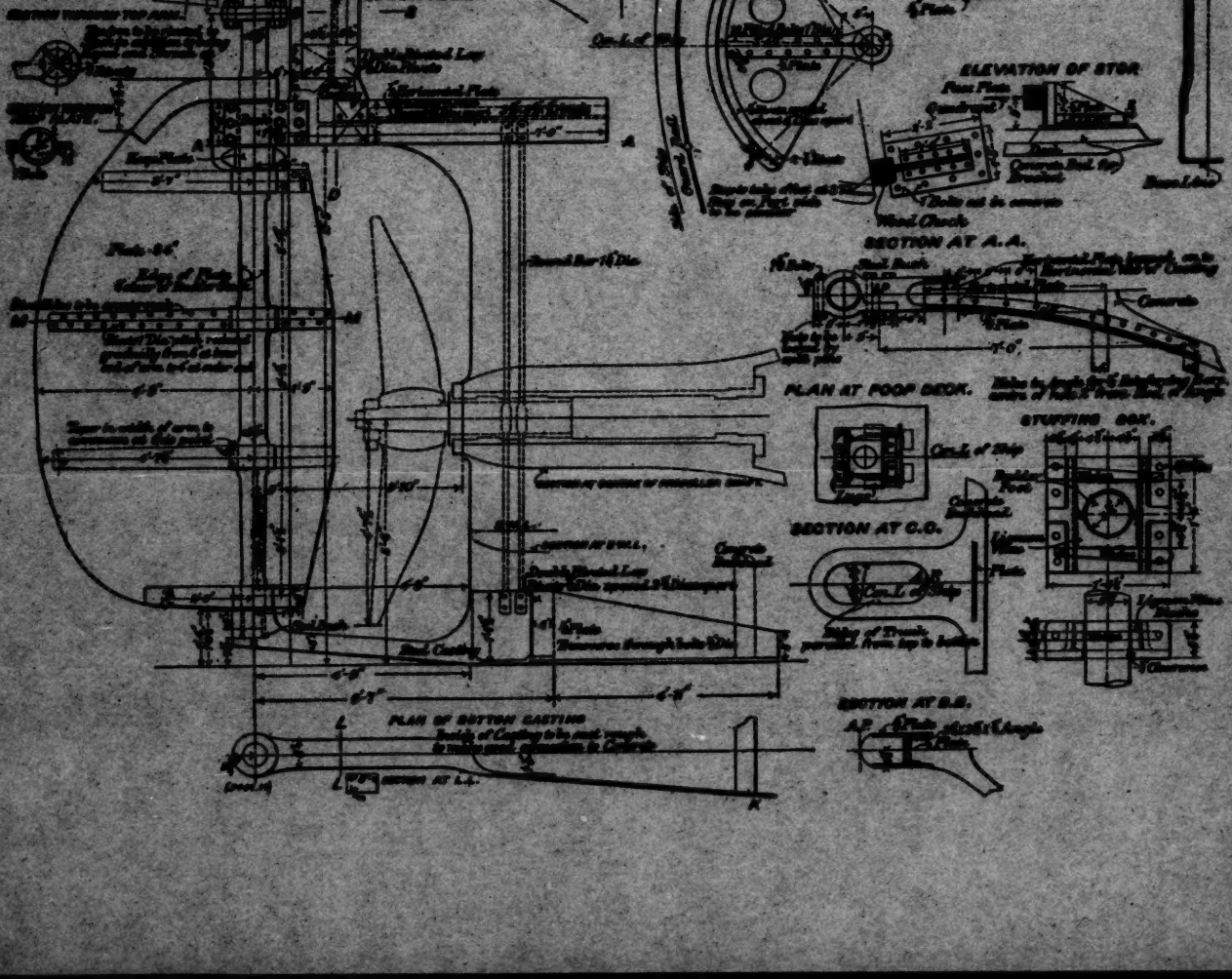


Fig. 16.
SECTION 5'-0" FORWARD OF A.P.
PLAN OF TILLER
SECTION AT D.D. (18" FORE OF A.P.)
SECTION AT M.M. THROUGH RUDDER ARM.
PLAN OF QUADRANT.
SECTION AT K.
ELEVATION OF STOP


ELEVATION OF STOR
SECTION AT A.A.
PLAN AT POOP DECK.
STUFFING BOX.
SECTION AT C.D.
SECTION AT E.B.
PLAN OF BOTTOM CASTING

steel ship, but that in building successive ships of the same size and form there will be a marked reduction owing to the possibility of using repeatedly a large proportion of the shuttering or molds. By this means I believe that, with all the material to hand, a vessel similar to that described in this paper could be completed in three and a half to four months.

I regret that neither time nor space will allow me to indicate other types under construction in the yard, all possessing interesting characteristics. When completed their performances will no doubt add a helpful contribution to the study of ferro-concrete construction as applied to shipbuilding. Neither can I touch on the question of the application of discard steel for constructional purposes, as largely advocated by some builders, nor on the future possibilities of ferro-concrete as applied to shipbuilding when the time arrives when we shall not be driven to its adoption by stress of war conditions.

I admit some limitations in the paper, due to absence of precise knowledge and experience of the sea tests of the ships, but I hope to have the honor on another occasion of completing this contribution by giving the service results of the vessels described, when opportunity will probably occur to deal with other points omitted from the present paper. In conclusion, I should like to express a word of thanks to those associated with me in this work: the members of the staff of the Ferro-Concrete Ship Construction Company, of Barrow-in-Furness, who have worked so energetically and untiringly in the design and construction of the vessels; and to the directors and staff of Messrs. L. G. Mouchel and Partners, who have brought great ability and experience to bear on the solution of some of the difficult problems involved; also to the directors of the Yorkshire Hennebique Company, who have by their experience shown us how, in many ways, the time of construction can be shortened.—"Engineering."

LUBRICATION OF AIR-COMPRESSOR CYLINDERS.

Recent disastrous explosions in air-compressor systems present striking examples of the danger existing from the use of ordinary engine oil in the air cylinders of air compressors. Only a pure mineral oil, with a flash point as high as good lubricating qualities will permit, should be used. As little as possible of even the best oil should be used.

Numerous cylinder oils are compounded, and such oils are likely to produce a carbon that will stick the valves and collect on valve faces and other parts of the cylinder and valve chambers, resulting in a dangerous condition.

Air receivers are liable to explosion from accumulated oil deposits. Every receiver should be equipped with a pressure gage, a safety valve and proper drains, and all reservoirs and likely places of deposit in the air line should be thoroughly and frequently drained and cleaned. It is bad practice to have the inlet of an air compressor take from a hot or dusty room—the air should be cool and as clean as possible.

The practice of throwing kerosene oil into the inlet of an air compressor to clean it is an extremely dangerous one, and the cause of an explosion under such circumstances is not difficult to understand. Lubrication of the air cylinder with soapsuds (preferably made of soft soap, about one part soap to fifteen parts water) for a few hours each week (or less frequently if the load is light), instead of oil, will help very materially in keeping the cylinder clean. The only danger from the use of soapsuds is rust, and this should be overcome by being careful to discard the soap and feed the cylinder with oil an hour or so before shutting down. The receiver blow-off should then be opened and the accumulation of oil and water drained off.

An air-compressor engine should not be controlled by the air pressure alone, as many are, but should be fitted with an auxiliary governor which will act as soon as the speed rises above a certain predetermined limit. This will prevent the engine from "racing" in case an accident to the tanks or piping causes a sudden lowering of the pressure. It is not necessary for an explosion to take place to produce a lowering of the pressure, as the giving way of a pipe, valve or tank from any cause will have the same effect.—"The National Safety Council."

BOOK REVIEWS.

New Publications: Finding and Stopping Waste in Modern Boiler Rooms, by Engineers of the Harrison Safety Boiler Works, Philadelphia, Pa.; cloth; 9 inches by 7 inches; pages, 274; 213 illustrations; $1.00.

The saving of coal is the purpose of this practical handbook. It is addressed to power-plant owners, managers, engineers, and firemen. It contains tables and charts, the results of experiments and tests, references being made wherever possible to the original authorities. The latter include many well-known engineers and technical authors. The United States Bureau of Mines bulletins on the utilization of fuel have been drawn upon freely.

The work is divided into five sections, the first of which deals with "Fuels." Here are considered kinds and classification of coals, coal sampling, proximate and ultimate analyses, heating value, ash and clinker, purchase under specification, washing, storage, weathering, and measurement of coal, oil fuel, and gaseous fuels.

The second section, entitled "Combustion," treats of the chemistry of combustion, air theoretically required, grates and grate surface, hand firing, mechanical stokers, furnace temperature, furnace gases, draft, flue and stack proportions, flue-gas analysis, CO_2 recorders, what CO_2 and CO indicate, excess air, smoke and smoke prevention, burning oil fuel, gaseous fuel, and powdered coal.

The third section, entitled "Heat Absorption," treats of heat transmission by conduction, convection, and radiation, heat transfer, air heaters and superheaters, improving heat absorption, relation between heating surface and boiler capacity, boiler setting, refractories, soot, scale, softening and heating feed water.

The fourth section, entitled "Boiler Efficiency and Boiler Testing," covers heat balance, heat absorbed by boiler, heat losses due to moisture in coal, hydrogen, stack gases, CO, combustible in ash, efficiencies, and boiler trials.

The fifth section, "Boiler Plant Proportioning and Management," discusses various arrangements of auxiliaries with regard to their effect upon feed heating, and also describes the Polakov functional system of boiler-room management.

The book was compiled by GEORGE H. GIBSON, Member American Society Mechanical Engineers, assisted by PERCY S. LYON. It is a fine compilation of very useful data on combustion.

THE CAMOUFLAGE.

The JOURNAL greets with enthusiasm the entrance of "The Camouflage" to the journalistic field. It is the official paper of the Key West Naval Training Station, published by the Naval Athletic Association, and devoted to "the interests of the personnel of all branches of the U. S. Service."

The editor has read with pleasure the opening numbers of this paper. It will do an excellent work by acquainting the American people with their Military Services.

May its propaganda be right, its life long and successful.

SCIENTIFIC MANAGEMENT APPLIED TO INSTRUCTION AND TRAINING IN FIELD ARTILLERY. By MAJOR WILLIAM E. DUNN, FIELD ARTILLERY, N. A. J. B. LIPPINCOTT COMPANY. 96 pp.; 75 cents net.

This handbook, written in 1916, applies the principles of industrial efficiency to the biggest business of the day. War is a business. Field artillery has played, and will continue to play, an important part in bringing the business to a successful conclusion.

Major Dunn's work is compact, easy to understand, written in attractive style, and should prove extremely useful to gunners—officers and men—who are training for efficient service.

The author does not claim originality in ideas. He certainly has compiled the ideas of others, in an original way, into a manual of interest and great service.

The paper and binding are serviceable; the type is clear. The make up of the work reflects credit upon the publisher.

JOSEPH PENNELL'S PICTURES OF WAR WORK IN AMERICA. J. B. LIPPINCOTT COMPANY, PHILADELPHIA. $2.00 net. XXXVI prints.

Reproductions of a series of lithographs of munition works made by him with the permission and authority of the United States Government, with notes and an introduction by the artist.

The sub-title does not fully indicate the scope of the work. Of the thirty-six lithographs—all of which are well reproduced—a goodly number are of ships and ship yards. Mr. Pennell has caught the spirit of the work and interpreted it as an artist, but as an artist who was, as well, an engineer. Some of the Italian masters were, and the war has produced a French school of artist-warriors whose work is well worth while. May it not be that this is an American member of the same schools?

The collection of prints is unusual. The press-work is good, the binding attractive.

The spirit of America, at work for the Allies, permeates the collection like the *deus ex machina* of the classic dramatists.

MANUAL OF RECRUIT INSTRUCTION FOR FIELD ARTILLERY. BY MAJOR WILLIAM E. DUNN, FIELD ARTILLERY, N. A. J. B. LIPPINCOTT COMPANY. 96 pp.; 75 cents net.

Major Dunn's last work shows a continuation of his compact, lucid style. Commanders and non-commissioned officers in the battery service will find it extremely valuable in training recruits. It presents skeleton outlines of intensive courses, based on Army Regulations, for educating new men in the

essentials of artillery practice in the field. Provision is made for lectures by the battery commander to recruits, with the intention of starting the men in the proper way and making them "Soldiers at Heart."

The binding, paper and type are uniform with his earlier work and indicate the publisher's usual care for details.

The Service should await Major Dunn's projected works with interest and may expect worth-while work.

THE SOLDIERS' ENGLISH AND ITALIAN CONVERSATION BOOK. By IDA DICKINSON. J. B. LIPPINCOTT COMPANY, Philadelphia, 728 pages, 75 cents net.

This small, handy phrase book is uniform with and translated and adapted from Walter M. Gallichan's English-French Conversation Book. It contains hundreds of useful sentences and words, enabling the American soldier at the front to converse with the Italian Allies. It is specially adapted to military use.

OFFENSIVE FIGHTING, by MAJOR DONALD M. MCRAE, M.C., 196 pages; 18 illustrations; J. B. LIPPINCOTT & Co., Philadelphia. Price, $2.00.

This book, written from the results of experience in the grim game of offensive fighting, is intended to put this experience in the hands of the men and officers to and including majors who must perform similar service. The chapter headings show the scope of the book.

I. Mine Crater Consolidation.
II. Trench Raid.
III. Patrols and the Control of No Man's Land.
IV. Front Line Observation.
V. Coöperation Between Infantry and Artillery.
VI. Infantry in Attack.
VII. Trench Routine.
VIII. Notes on Use of Prismatic Compass.

MECHANICAL ENGINEERS' HANDBOOK, by LIONEL S. MARKS, B. Sc., M. M. E., Editor-in-Chief, and Staff of 50 Specialists. Published by MCGRAW-HILL BOOK CO., New York, N. Y. Limp leather; 1836 pp. Price, $5.00.

This Handbook presents a marked departure from the old type handbook which, generally speaking, is a compilation by one author of existing data, in that it is the coöperative effort of nearly fifty specialists in the various branches of Mechanical Engineering. The section of "Properties of Engineering Materials" is alone the work of thirteen specialists. All this material has been ably edited and arranged by the Editor-in-Chief to form the premier of its field.

The list of contributors includes the most eminent men in their respective fields of Engineering. This book should be on every engineer's shelf.

SUBMARINES.

A LIST OF REFERENCES IN THE NEW YORK PUBLIC LIBRARY.

This is a very interesting compilation of books, magazines and Technical Papers relating to SUBMARINES. It is compiled by Mary Ethel Jackson. In a foreword by Simon Lake it is stated, "Much of the literature relating to submarines is based on hearsay, romance and imagination, but imagination may lead to scientific development, so that both the student of the romance of 'the mysteries of the deep' and the investigator of its use as a weapon of war, or its possibilities in the commercial field, will appreciate the labor which has been put into the preparation of this bibliography."

OBITUARY.

JOHN D. FORD, REAR ADMIRAL, U. S. NAVY.

John D. Ford was born in Maryland in 1840, and received his early education in the public schools in Baltimore. He attended the Maryland Institute of Design, and was graduated, taking the first Peabody prize. He took a post-graduate course at the Potts School of Engineering, from which he was graduated in 1862.

He entered the Navy as a Third Assistant Engineer in July, 1862, and was at once assigned to duty on board the *Richmond* in the West Gulf Blockading Squadron. He was in the *Richmond* at the battles of Baton Rouge and Port Hudson. He commanded a battery of four IX-inch guns (guns taken from the *Richmond*) with a crew of mechanics from the ships, in the west wing of the Army, in the siege of Port Hudson. He made the port shutters, of plate iron, for the protection of his men against the sharpshooters of the enemy, and he made the plate-iron armor for the military tops of the *Richmond*, which were believed to be the first ever used for the purpose.

After the fall of Port Hudson he resumed duty on board the *Richmond*, in the blockade off the Bay of Mobile. Promoted to Second Assistant Engineer in February, 1864, he was ordered to the prize steamer *Donegal*, to take her home, but he returned at once to the *Richmond*. He was in the *Richmond* at the battle of Mobile Bay, and took part in the destruction of the gunboats *Gaines*, *Morgan* and *McRae*, and in the capture of the *Selma* and *Tennessee;* and also in the reduction of Forts Morgan and Powel. He was temporarily assigned as Chief Engineer of the prize *Tennessee*, but later to the *Selma*, which vessel was employed in removing the obstructions in front of the city. He was assigned to the *Arizona*, which vessel was destroyed in the Mississippi River, but the crew was saved by swimming. He was then assigned as Chief Engineer of the

Estrella, and later to the *Sebago*, in which latter he served until the end of the civil war.

He was on experimental duty on board the *Pensacola*, at Baltimore, in 1865–6, and was in the *Sacramento* when she was wrecked on the coast of India in 1867. He reached home in a British bark in November, 1867, and was assigned to duty in the Norfolk Navy Yard.

He was promoted to First Assistant Engineer in 1868 and ordered to duty on board the *Swatara*. He served on the *Miantonomoh* in 1871, the *Hartford* in 1872, on the China Station. He afterwards served in the *Tennessee* on the same Station, and, in 1884, in the Bureau of Steam Engineering.

He was then detailed to organize and take charge of a class in Mechanical Engineering at the Manual Training School in Baltimore, where he remained six years (as long as the Department could spare an officer). His work there was eminently successful, and the college still continues the course on the lines laid down by Mr. Ford.

He then served on board the *Alert*, in the Behring Sea surveys, and was promoted to the rank of Chief Engineer in 1890, and ordered to the Examining Board.

He inaugurated the class in Mechanical Engineering in the Maryland Agricultural and Mechanical College in 1894, and served as professor for two years. He superintended the construction of the machinery for the cruiser *Brooklyn* (Schley's flag ship), and served on board that ship as Chief Engineer. He joined the Asiatic Fleet in 1898, and served as Fleet Engineer on Dewey's flag ship, the *Olympia*, during the destruction of the Spanish fleet at Manila Bay.

He was transferred into the line of the Navy, as Commander, in 1899, and ordered home from Manila. Promoted to Captain in February, 1902. His last duty was an Inspector of Ordnance at Sparrows Point, where he was serving when retired. He was retired on reaching the age of 62 years, but, by an Act of Congress, he was promoted to the grade of Rear Admiral upon retiring.

Rear Admiral Ford was resourceful and, being a practical mechanic and draughtsman, he had little trouble in executing his designs. He had a fortunate faculty for stating a proposition and for making his ideas clear. He was a natural organizer, which is essential to inducing a following. He had a nice way of setting his subordinates right without irritation or offense. He was a jolly good shipmate, and contributed much to the pleasure of his shipmates.

The American Society of Naval Engineers, the Military Order of the Loyal Legion, the Associated Veterans of Farragut's Fleet, the Society of Mechanical Engineers, and the Protestant Episcopal Church numbered him among their members, and he was a Royal Arch Mason.

He was buried in Greenmount Cemetery, in Baltimore, on the eleventh of April, 1918, services being held at Saint Michael's and All Angels' P. E. Church, and his body committed to the ground with the Masonic services of Union Lodge, No. 60, of which he had been a member for more than half a century.

He is survived by his widow, four sons and one daughter.

G. W. B.

JOHN ADDISON BAXTER SMITH.

John Addison Baxter Smith, one of the oldest members of the American Society of Naval Engineers, died in New Orleans in March, 1918, while on a visit to his daughter.

He was born in Baltimore in 1843, where he was educated, and was graduated from the Maryland Institute. He entered the Navy during the Civil War, in April, 1863, the examination being competitive. He graduated number 5 in a class of 13, and was warranted a Third Assistant Engineer.

He was ordered at once to the steam sloop *Housatonic*, then blockading off the coast of South Carolina, which vessel was torpedoed and destroyed in February, 1864. He was rescued by a boat from the *Canandaigua*, was assigned to duty on

board the steam frigate *Wabash*, and afterwards the gunboat *Paul Jones*; later he was transferred to the *Monongo*, promoted to Second Assistant Engineer, and made the voyage to San Francisco in that vessel.

In 1867 he was employed on the extended trial trips of the *Moshulu*, *Pushmataha*, *Contoocook* and *Miantonomoh*. There was, at that time, no appropriations available to defray the cost of experiments, but advantage was taken of the trial trips to record all possible information and data, the results of which enabled the Bureau of Steam Engineering to determine and report many important discoveries.

In 1867–8 Mr. Smith served on board the *Saco* (an experimental ship) and later on the *Seminole*, and was on that vessel during her voyage in pursuit of the famous pirate *Telegraph*. He served on the *Mayflower* in the famous Shufeldt expedition, surveying for a canal route via Tehuantepec, and, later, at the League Island Station, as it was then called.

In 1873 he served on board the monitor *Saugus*, during the "Virginius excitement." He was promoted to Passed Assistant Engineer in January, 1873, and ordered to the Pensacola Navy Yard as Chief Engineer, where he remained during the yellow fever epidemic at that place. Later he served on board the monitor *Wyandotte*, the steam sloop *Hartford*, the *Tallapoosa*, and at the Norfolk Navy Yard.

He served on board the *Atlanta* (the first of the new, steel, Navy) during her extended trial trips. In 1888 he was a member of the Advisory Board, in Washington, and in 1889 became Inspecting Engineer at the Cramps Ship Yard at Philadelphia, where the *Baltimore* and *Philadelphia* were being built. He was promoted to Chief Engineer, February, 1892, and ordered to the *Montgomery*, where he served two years, when he was transferred to the battleship *Texas*. He was Chief Engineer of the Norfolk Navy Yard in 1896–7–8, and in 1899 was transferred into the line of the Navy as a Commander, when he was ordered to the Navy Yard, New York, where he

did the very important work of rehabilitating the steam engineering plant, changing from the shaft-and-pulley-driven tools to the direct electric drive, using alternating current of high voltage with a motor to each machine tool, in the (then) largest steam-engineering plant in the country. It is not too much to say that in this feat he was a pioneer. The plant was, at once, a great success. He was promoted to Captain on the 8th of June, 1902, and in 1904 was made General Inspector for all outside work under the Bureau of Steam Engineering.

He was retired, at the age of 62, on the 21st of March, 1905, with the rank of Rear Admiral, having to his credit 15 years and 11 months of sea service and 22 years of shore duty.

Rear Admiral Smith was a large, powerful man, capable of great endurance, and always ready for duty. He was an agreeable and popular shipmate, always frank and outspoken, but always considerate of the rights and feelings of others.

He was very domestic in his habits; a devoted husband and father, whose pleasure was to lighten the burden of his family and bring sunshine into their lives.

He is survived by a widow, two daughters (Mrs Freyer, who married Major Freyer of the Marine Corps, and Mrs. Ramsay) and by one son, Paymaster J. A. B. Smith of the Navy.—G. W. B.

ASSOCIATION NOTES.

The following members and associates have joined the Society since the publication of the February, 1918, JOURNAL:

MEMBERS.

Clement, Emory F., Lieutenant Commander, U. S. N.
Hansel, Cyrus S., Ensign (T.), U. S. N.
Heiner, John N., Lieutenant, Coast Guard.
Kalde, Frederick R., Ensign (T.), U. S. N.
Rung, H. E., Ensign, U. S. N.
Skinner, A. C., Lieutenant, U. S. N.
Torbet, Mayson W., Lieutenant, Coast Guard.
Troll, Walter M., Lieutenant, Coast Guard.
Waldo, G. W., Lieutenant, U. S. N.

ASSOCIATES.

Adae, C. F., Lieutenant, R. F.
Adams, John, Jr., Ensign, R. F.
Anderson, J. T. H., Ensign, R. F.
Barry, E. H., Ensign, R. F.
Borden, John, Lieutenant Commander, R. F.
Bourne, P. O., Chief Engineer, Blake and Knowles Works, Worthington Pump & Machinery Corp., Cambridge, Mass.
Bridges, J. W., Ensign, R. F.
Burbank, A. G., Lieutenant, R. F.
Burgess, L. B., Lieutenant, R. F.
Carse, H. E., Instructor in Mathematics and Engineering, N. R. Training Camp, San Pedro, Cal.
Cooper, William T., Lieutenant, R. F.

Damuth, W. C., 238 Lafayette Avenue, Brooklyn, N. Y.
Diehl, George S., Lieutenant, R. F.
Ferguson, William, Ensign, R. F.
Gray, John W., Room 410, Harrison Building, Philadelphia, Pa.
Griscom, George S., Jr., Engineer, 11 Rumford St., Liverpool England.
Hay, W. O., Lieutenant, R. F.
Hechler, F. G., M. E., Engineering Experiment Station, Annapolis, Md.
Kerlin, Ward Dix, Secretary-Treasurer Camden Forge Co., Camden, N. J.
Perkins, Donald, 263 Arlington Avenue, Jersey City, N. J.
Sinclair, S. W., Lieutenant, R. F.
Wheeler, O. J., Lieutenant, R. F.